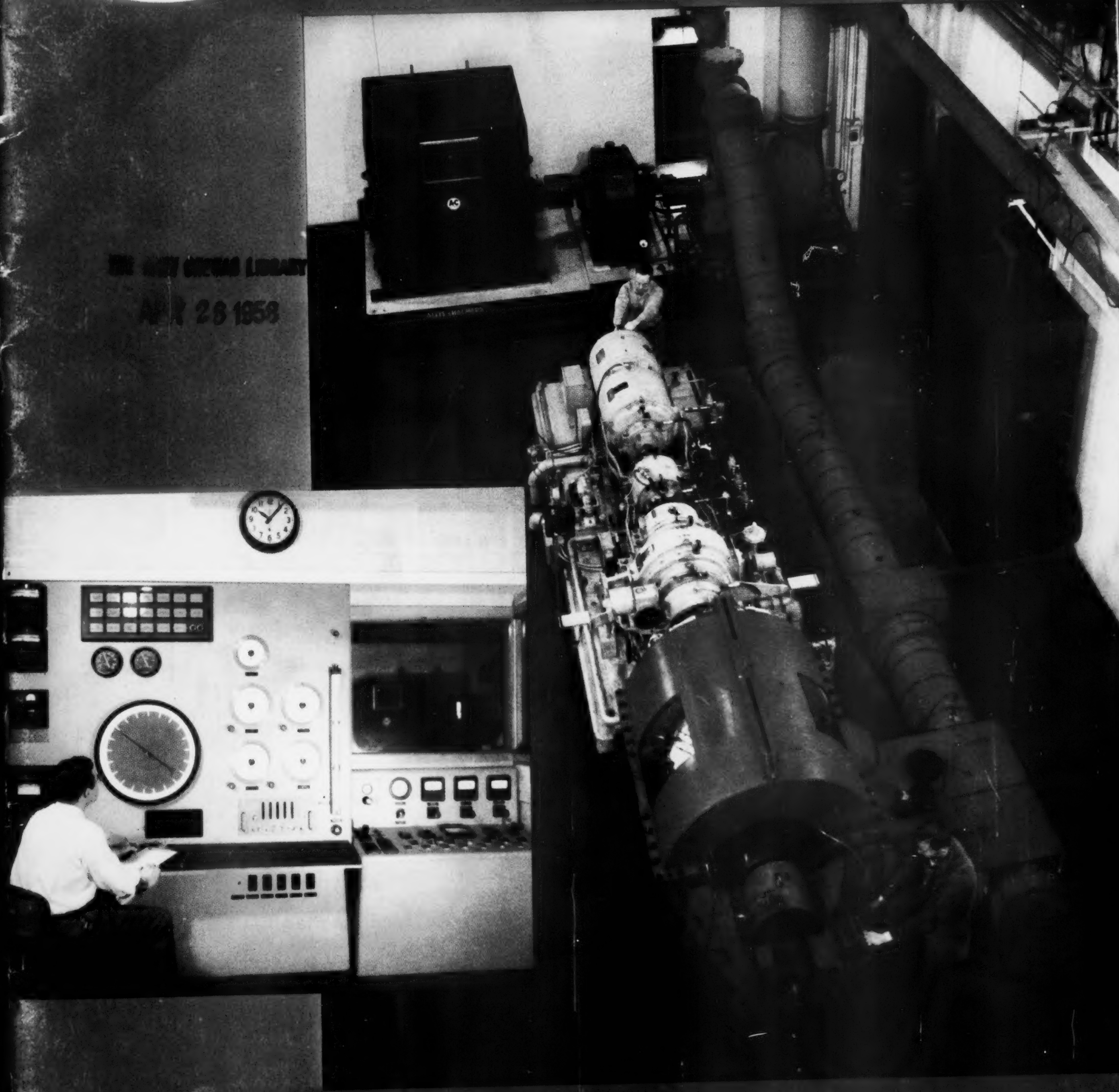
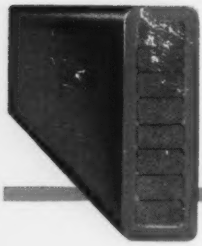
ALLIS-CHALMERS Electrical REVIEW

1st QUARTER 1958





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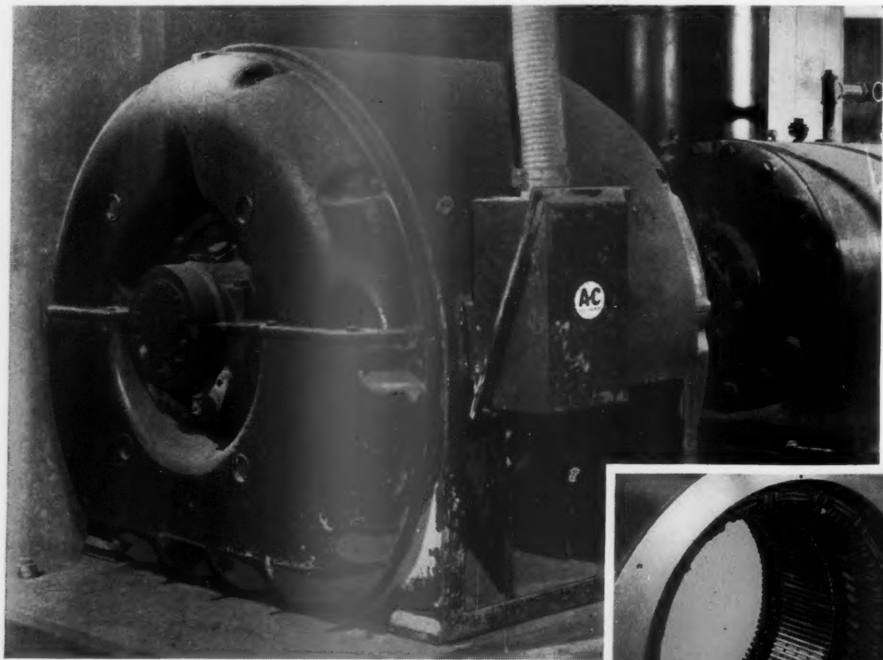

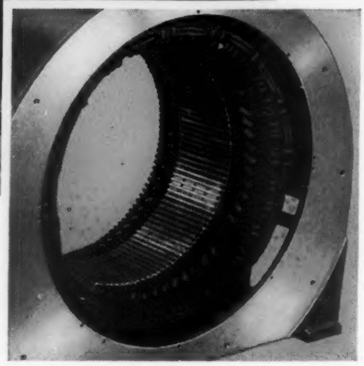

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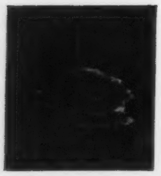

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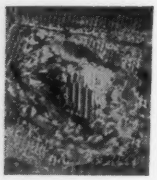

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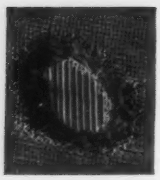

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Electrical REVIEW

THE COVER

NEVER-ENDING QUEST for even greater steam turbine efficiencies is pursued in Allis-Chalmers new Experimental Turbine Laboratory. Here air is used instead of steam to test various turbine blades in the elaborately equipped high precision laboratory described on page 10.

Allis-Chalmers Staff Photo by Mike Durante

Allis-Chalmers

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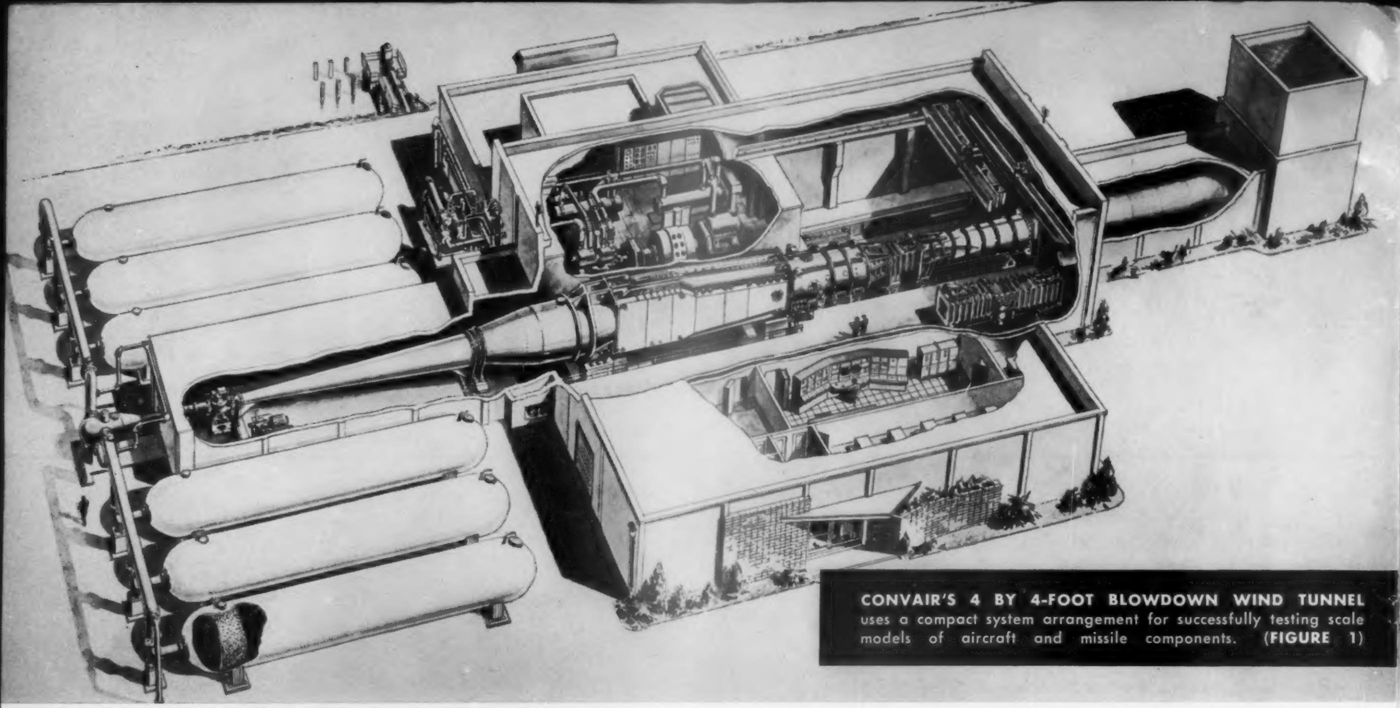


CONVAIR'S 4 BY 4-FOOT BLOWDOWN WIND TUNNEL uses a compact system arrangement for successfully testing scale models of aircraft and missile components. **(FIGURE 1)**

CONVAIR BLOWDOWN WIND TUNNEL . . .

Develops Transonic and Supersonic Velocities

by **R. E. CLAUDE**
Compressor Department
and
M. F. GAY
Electrical Application Department
Allis-Chalmers Mfg. Co.

New wind tunnel developing air speeds of up to 3800 mph will accelerate research in supersonic flight to greatly further modern aircraft and missile development.

SOMETIME IN THE EARLY PART OF 1958, a valve will be opened and air at 3800 mph will speed through the 4 by 4-ft test section of the Convair Division of General Dynamics High Speed Wind Tunnel at San Diego, California. This tunnel, designed by the Fluidyne Engineering Corp., built by the Chicago Bridge and Iron Company and supplied with air from a special combination of compressors, motor, switchgear and controls, will be the first operating blowdown wind tunnel of its type in this country.

Many thousands of hours of wind-tunnel testing are necessary during the development of today's modern multi-million dollar supersonic aircraft or missiles, to prove that they will equal or exceed their predicted performance when they become air-borne for the first time. Faulty or unproven design at any stage of progress cannot be tolerated because of the tremendous investment in time, engineering, and cost of prototypes. Particularly, a pilot's life could be jeopardized if he were to operate unsafe equipment that had not been tested. However, this great amount of wind-tunnel testing time, particularly at the transonic (Mach 0.7 to 1.3) and supersonic range (Mach 1.3 to 5), is not easy to obtain when needed because of the lack of these types of tunnels. Because the present work load for these few tunnels is so great, the aircraft companies have found it advantageous to build their own tunnels for the research necessary to remain competitive.

The blowdown-type wind tunnel, while not as preferable as a continuous-type tunnel, is the answer to the needs of the aircraft industry. The necessary research data can be gathered with only a fraction of the installed horsepower and over-all facility cost required for a continuous flow-type tunnel.

Three compressors power blowdown tunnel

The high velocities necessary to obtain supersonic speeds are generated by compressing air to 645 psia through three stages of centrifugal compressors shown in the cutaway of the power room in the top center of Figure 1. This high pressure air is stored in the six tanks on the left at 600 psia. In this tunnel, pumping the tanks from atmospheric pressure to the 600 psia takes approximately 30 to 40 minutes. To stabilize the temperature of this high pressure air, beds of alumina balls are used in the storage tanks to keep the air within the desired limits.

When it is desired to run a test, an electronically controlled quick-opening valve, shown between the tanks in the left center of Figure 1, is opened, allowing the high pressure air to enter the settling chamber shown where the tunnel enters the building. This large chamber with several layers of screening straightens the flow so it can enter the flexible nozzle of the tunnel with a minimum of turbulence.

The flexible nozzle (the box section between the power room and control room) determines the rate of supersonic flow. The sides of the nozzle are fixed, but the top and bottom plates are flexible and actuated by hydraulic jacks to give the proper opening and curvature necessary for the desired Mach number.

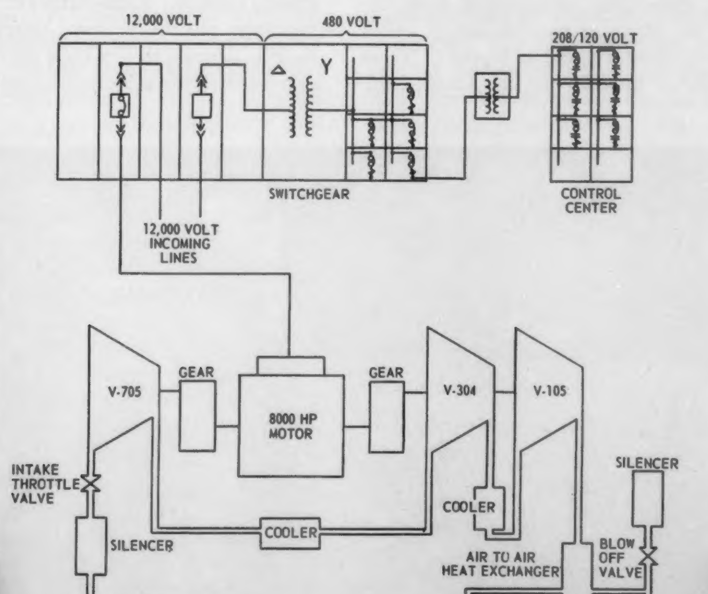
The air then flows through the test section where the model is mounted on a sting support which will move the model to the desired positions during the run. The data obtained during the test run on pressures, temperatures and forces at various locations on the model is fed directly into computers and translated into the desired parameters. There are two different types of test sections, one for supersonic testing and one to cover the transonic range. Changing either of the sections takes but a few hours.

From the test section, the air travels through a variable diffuser, into a fixed diffuser and through a silencer (top right of Figure 1) to the atmosphere. The variable diffuser, like the flexible nozzle, has movable sidewalls actuated by hydraulic jacks. The purpose of these sections, in addition to increasing the tunnel efficiency, is to allow the supersonic air to again become subsonic before it enters the silencer and exhausts to the atmosphere.

The test times are short, but of sufficient duration to obtain valid data. Only 20 to 60 seconds are usable before the tanks reach too low a pressure for further operation, depending upon the desired speed. Two to four runs per hour are available, again depending on test requirements.

Equipment selection rests on cooperation

In a compressor and electrical drive system, the basic consideration is the furnishing of a given volume of air at a particular pressure and temperature. These considerations establish the type of compressor or a combination of compressors that is most ideally suited for those requirements. However, in the selection of the compressor plant, attention must also be given to the types of drive motors and speed-increasing gears that could be used. Proper arrangement of these basic components results in a combination of commercially available equipment, for an over-all installation with very good performance characteristics and long, trouble-free life. Continual and careful cooperation



SIMPLIFIED diagram contains functional layout of electrical system, compressor plant and tunnel circuit. **(FIGURE 2)**

of those responsible for the compressor application and for the selection of the electrical system for this integrated blowdown wind tunnel is vitally important for the successful conclusion of the project. The voltage characteristics and interrupting capacity of the power system on the sudden application of a large motor must be examined. Other limitations that may exist in the power system, as well as protection for the equipment, interlocking, instrumentation and operating controls, must also be studied.

Compressor arrangement develops 645 psia

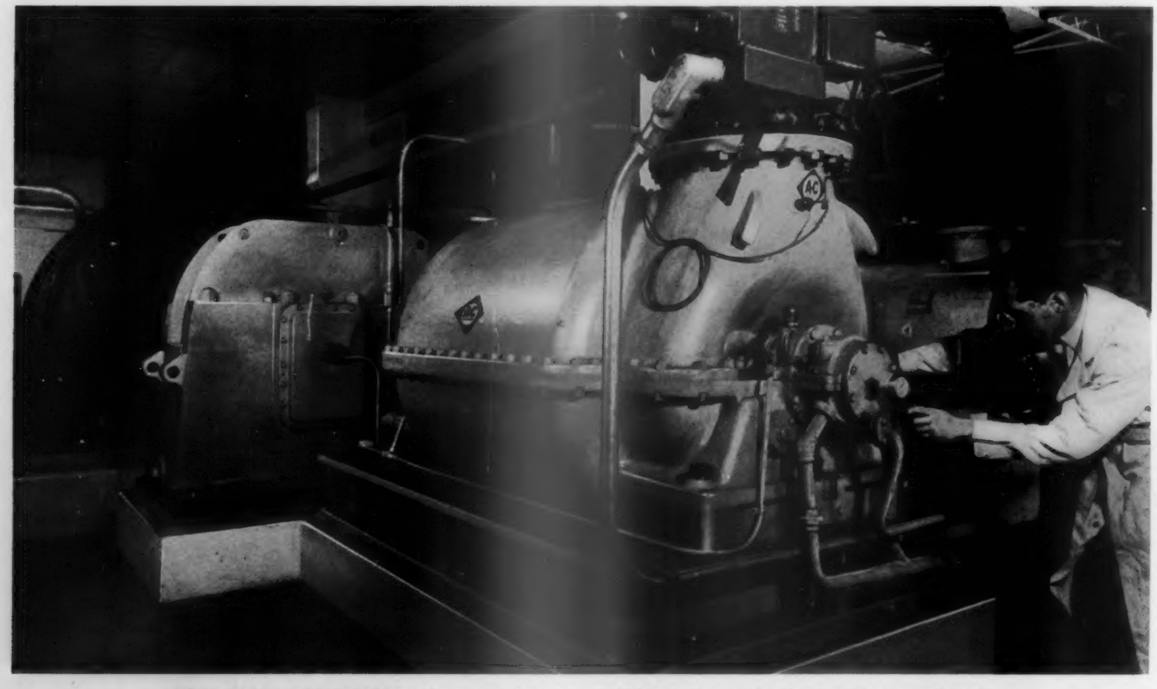
The compressor system is rated at 19,000 cfm at 68° F and 13.8 psia inlet conditions to obtain a tank pump-up time of approximately one-half hour. This is a compromise value, since the shorter the pump-up time, the larger the compressors and the installed horsepower. Centrifugal compressors were the logical choice because oil-free air is essential and the inlet volume was large enough to favor this type of equipment. It was also desirable to keep as close to standard machines as possible.

Since sea water from San Diego Bay is the cooling water supply, uncooled compressors are a definite advantage, with easy to clean intercoolers between each of the machines. Consequently, three compressor stages in series develop the necessary pressure ratio to get the air from the 13.8 psia inlet conditions to a discharge pressure of 645 psia. The additional 45 psia above the 600 psia tank pressure is necessary to overcome the pressure drops of the air-to-air heat exchanger, aftercooler, refrigeration dryer, the circuit back through the air-to-air heat exchanger, check valve and back pressure valve, as shown in Figure 2. These components are connected between the third-stage compressor and the storage tanks. Compressor pressure is relieved to atmosphere by the blowoff valve and silencer.

Both the second and third-stage compressor are run at the same speed because of their small size and the relationship of their ratios and number of impeller wheels. Speed-increasing gears were necessary, since it is impossible to run any of the compressors at a synchronous motor speed. Putting the drive motor between the first and second compressor seemed to be the best plan. Each gear could then be sized for minimum horsepower to be transmitted, resulting in a savings in initial cost. Figure 3 shows the first-stage centrifugal compressor, V-705, rated 19,000 cfm and 6756 rpm with a 4.5 pressure ratio, and its gear. Figure 4 shows the second and third-stage centrifugal compressors and their gear. The second stage V-304 is rated at 4325 cfm with a 3.59 pressure ratio, and the third stage V-105 is rated 1220 cfm with a 2.93 pressure ratio, both turning at 10,700 rpm. The V-105 compressor, although looking externally like the V-304, has a cast-steel casing because of the high discharge pressure.

Electrical system supplied by two lines

The electrical system for the Convair wind tunnel is a completely integrated system. The switchgear, main drive motor, auxiliary motors, auxiliary power and miscellaneous small motor starters are furnished complete with suitable interlocking, protective relaying, instrumentation, and operating controls. These components comply with Convair's operating procedure, power system limitations and equipment protection to provide a system that is easy to operate and requires the minimum amount of maintenance.



FIRST STAGE COMPRESSOR takes atmospheric air and delivers it at 62 psia through intercooler to inlet of second stage machine. Low speed gear directly couples first stage unit to motor. **(FIGURE 3)**

Figure 2 is a combination of a simplified single-line diagram of the electrical system, block diagram of the main components of the equipment, and flow diagram of the compressor plant with its associated piping and valves. Observe that there are two incoming 12,000-volt lines. One line was supplied by the power company for the large synchronous motor and is backed up by its own transformer. The other line is for supplying auxiliary power. Two lines are necessary because of the power system limitations. The motor for this wind tunnel is an unusually large load for this one part of the power company's system.

Single motor drives compressors

The motor is an 8000-hp, 12,000-volt, 1200-rpm, unity power factor, open self-ventilated synchronous motor. The motor has Class B insulation with a 60° C rise in a 40° C ambient on continuous load. The motor shown in Figure 5 has two pedestal-type sleeve bearings which are force lubricated from a common lubrication system that serves the compressors, gear and motor. Bearing temperature relays with high temperature alarm contacts are installed on each bearing. Six 10-ohm stator temperature detectors indicate motor operating temperature, and space heaters are provided to prevent the formation of condensation on windings and core when the motor is not operating.

The three primary motor leads and three neutral leads are brought out into a special terminal box. The terminal box is ventilated from the motor ventilating system. Three current transformers, mounted in the terminal box, are connected in the neutral of the motor winding and are used in the motor differential protection scheme. The box also provides space for making up the incoming cable with stress cones. Terminals are also provided for connection of the leads to surge and lightning protection equipment.

Since speed-increasing gears are coupled to each end of the motor, the flexible gear-type couplings between the motor and the gears are electrically insulated, and bearing pedestal insulation is provided for each of the bearing pedestals. This insulation prevents the flow of any shaft currents. One of the pedestals is normally grounded across its bearing pedestal insulation with a grounding strap. The removal of the grounding strap provides a convenient method of checking the insulation of the bearing pedestals and the flexible couplings.

Motor accelerates large inertia

When the 8000-hp synchronous motor is started it must accelerate its own inertia, the inertia of the rotating mass connected to it and supply the necessary power required to drive the compressor load. Motor starting must not produce objectionable voltage fluctuations at other loads that are supplied by the same power system. The 1200-rpm salient-pole, unity power factor synchronous motor accelerates an unusually large inertia. A specially designed amortisseur winding in the faces of the field poles has enough thermal capacity to absorb the energy required to accelerate this inertia. In accelerating from standstill to full speed, it is necessary to dissipate as heat in the amortisseur winding an energy equivalent to the kinetic energy

BOTH THE SECOND AND THIRD STAGE compressors rotate at the same speed. Third stage delivers 645 psia. **(FIGURE 4)**

of the rotating mass when it has reached full speed. The compressors, speed-increasing gears and motor itself have a minimum amount of inertia and the starting winding is carefully designed to handle that inertia.

The 8000-hp synchronous motor is started across the line without any objectionable voltage fluctuations on other loads in the power system. A separate power transformer supplies only the motor load at the 12,000-volt secondary voltage. The voltage drop through the separate transformer due to transformer impedance and motor-starting current, together with the voltage drop of the cable between the transformer secondary and the motor, causes an appreciable voltage drop at the motor terminals on starting. This voltage drop results in a decrease in the starting kva and starting current imposed upon the power system. The voltage drop at the motor terminals on starting is slightly more than 30 percent. This voltage drop makes it possible to start the motor without reduced-voltage starting equipment and the additional necessary switchgear. The lowered voltage causes reduced motor torques on starting, during acceleration and synchronizing, but with the type of load imposed by the compressors the motor torques are ample.

The motor accelerates and synchronizes in approximately 25 seconds. The initial voltage drop of slightly more than 30 percent remains fairly constant for approximately 18 seconds and then decreases during the latter part of the accelerating period. In starting, the motor input increases suddenly to approximately 2000 kw upon application of voltage and gradually increases to approximately 4200 kw during the accelerating period. After synchronizing, the input decreases to 650 kw, which includes the compressor load with the inlet throttle valve closed, the losses in the speed-increasing gear and in the synchronous motor.

Single line-up contains all control

The high and low voltage switchgear, control, instrumentation, and unit substation for the installation are contained in one single line-up of metal-clad cubicles. This single

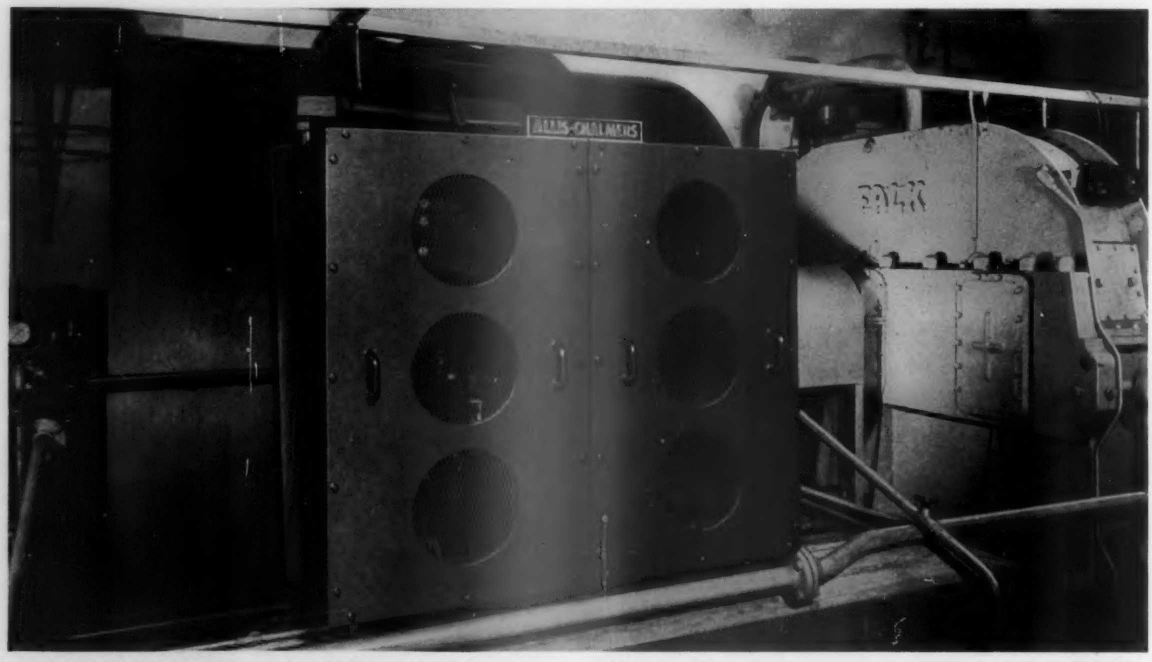
location contains the necessary instrumentation to show the performance and operation of the three compressor stages, the control for starting the main drive motor, and also furnishes 440-volt power to the larger auxiliaries and to a motor control center for smaller auxiliaries. The cubicles also include the necessary control switches, instrumentation and protective devices. Complete interlocking for the operation and indication of all the operating conditions of the electrical systems and the protection of the compressors, gears, main motor and electrical systems is also incorporated. The resulting cubicle line-up includes equipment that accomplishes much more than is the usual function of metal-clad switchgear. Most of the indicating lights for the indication of operating conditions are duplicated on the tunnel operator's control console for his observation. Figure 6 shows this line-up of equipment as well as 208/120-volt control center on the left end. There are eight units with a uniform height of 92 inches and a total length of 296 inches.

The first unit contains instrumentation for the compressor and some of the drive system. This includes gages for the indication of inlet and discharge temperature and inlet and discharge pressure of each of the three compressors. There is also a 20-point temperature recorder which includes high temperature alarm contacts with a point for each bearing of the compressors, gears, and drive motor. A temperature indicator with alarm contact and selector switch is mounted on the panel and is connected to the resistance-type temperature detectors embedded in the stator winding of the synchronous drive motor. Indicating lights identify bearings for high temperature, low lube oil pressure, excitation m-g set running, lube oil pumps running and the normal starting position of each of three air valves. A master control switch to start various auxiliary motors and an inlet throttle valve positioner are included on the panel.

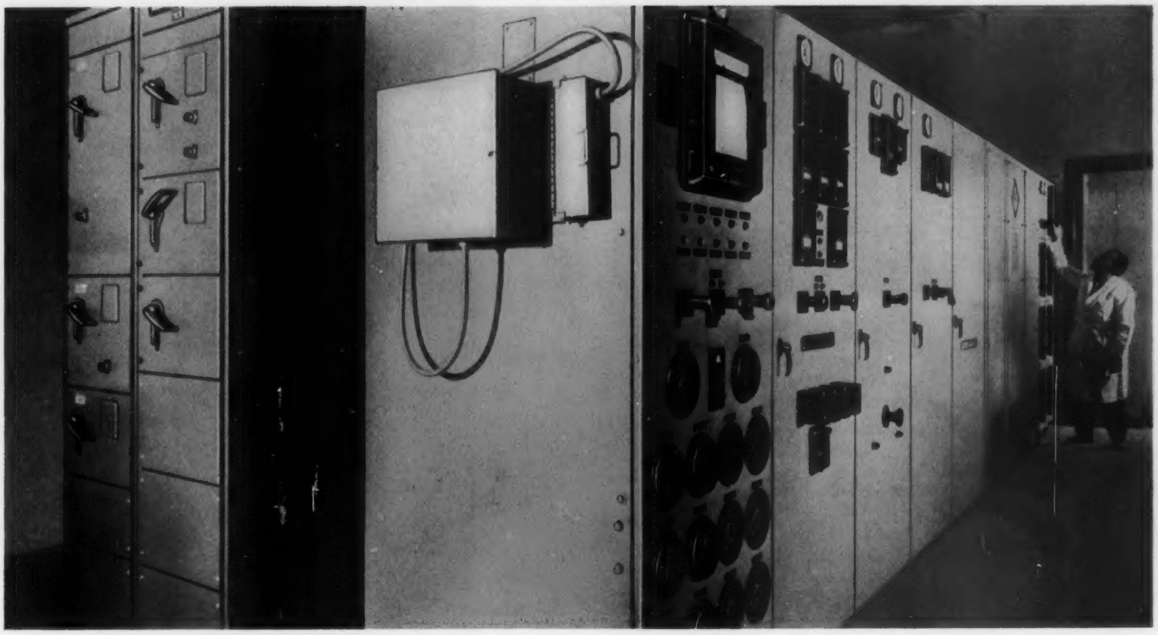
The second unit contains the main motor breaker and additional instrumentation, relays and control switches that are primarily associated with the main motor. The breaker is a 250,000-kva interrupting capacity, 1200-ampere air circuit breaker. The ammeter and varmeter are located on this unit. The differential, instantaneous, inverse-time and thermal overcurrent relays for motor protection are located directly beneath the meters. An emergency stop pushbutton, breaker control switch, approval light for starting and a cage winding high temperature light plus auxiliary relays are also mounted on this panel. Additional control items are mounted inside.

The number three unit is the incoming line and field application controls for the main motor. An ac voltmeter, field excitation ammeter, auxiliary relays, phase sequence and undervoltage relay, and exciter field rheostat are mounted on this unit.

The remaining units in the switchgear line-up consist of a 750-kva, 12,000/480-volt substation for auxiliary power. There is an incoming line unit supplied from the transformer for auxiliary power supply. The breaker unit has a 250,000-kva interrupting capacity, 1200-ampere air-magnetic circuit breaker, with an ammeter, overcurrent relays and breaker control switch mounted on the panel. The next two units are a transition unit and a unit enclosing a delta-to-ground wye dry-type transformer rated 750 kva, three phase, 60 cycles. The final two units make up a group of low voltage feeders for 440-volt auxiliary power. In these two units there are eight compartments, including current and potential transformer compartments,



SYNCHRONOUS MOTOR, rated at 8000 hp, drives all three compressors. Large terminal box, containing current transformers and cable connections, provides necessary clearance for the 12,000-volt lines. The high speed gear is directly connected to the motor. **(FIG. 5)**



MOTOR CONTROL CENTER, 12-kv switchgear, 480-volt radial substation, operating controls and necessary protective equipment are integrated in a single metal-clad switchgear line-up. **(FIGURE 6)**

a blank compartment, two 400-ampere feeders, two 300-ampere feeders, and one 200-ampere feeder. The feeders supply power to various loads associated with the tunnel and to a motor control center through a 400 to 208/120-volt transformer. The control center furnishes power to the excitation m-g set drive motor, the hydraulic pump motor, the lube oil heater, the lube oil pump motor, the motor space heaters, and for building lighting.

System control is fundamental

Since it was desired to keep the compressor control fundamental but as automatic as possible, a very simple yet effective control system is used. It consists of an inlet throttle valve, a blowoff valve and a back pressure valve with a check valve, and individual mechanical pop-off pressure relief valves for safety purposes. The purpose of the control system is threefold: (1) to provide unloaded starting for the motor, (2) pump up the tanks to 600 psia and (3) provide short-time and long-term idling without actual shutdown. The power company servicing the installation could not allow a large 8000-hp motor to start and stop more than once or twice a day without disturbing their regular industrial and residential service.

To start the system the suction throttle valve is put into an almost closed position. This automatically locks open the blowoff valve through a limit switch while the back pressure valve is held closed. When the motor is up to synchronous speed and with the compressors at their operating temperatures, tank filling is ready to begin. The inlet throttle valve is slowly opened and the compressor system is brought up to slightly above operating pressure and blows off through the blowoff valve and silencer to the atmosphere. The blowoff valve switches from locked open to automatically open at this time by the action of a second limit switch on the inlet throttle valve. The system is now in short-time idling and ready to start pumping.

To start the tank-filling operation, a switch is turned on the panel in the control room or power room, causing the back pressure valve to go on automatic operation to meter the 600 psia air into the tanks. The blowoff valve will automatically close because the system's pressure has fallen below its setting. The air pressure into the tanks must remain at 600 psia, since the refrigerator dryer is only efficient at this pressure and dry air is an absolute must to prevent condensation (with attendant condensation shocks and possible mechanical damage) within the wind tunnel during the high velocity runs.

When the tanks reach 600 psia, a pressure switch on the tank causes the back pressure valve to close, thereby building up the system back pressure to open the blowoff valve and put the compressor system into short-time idling.

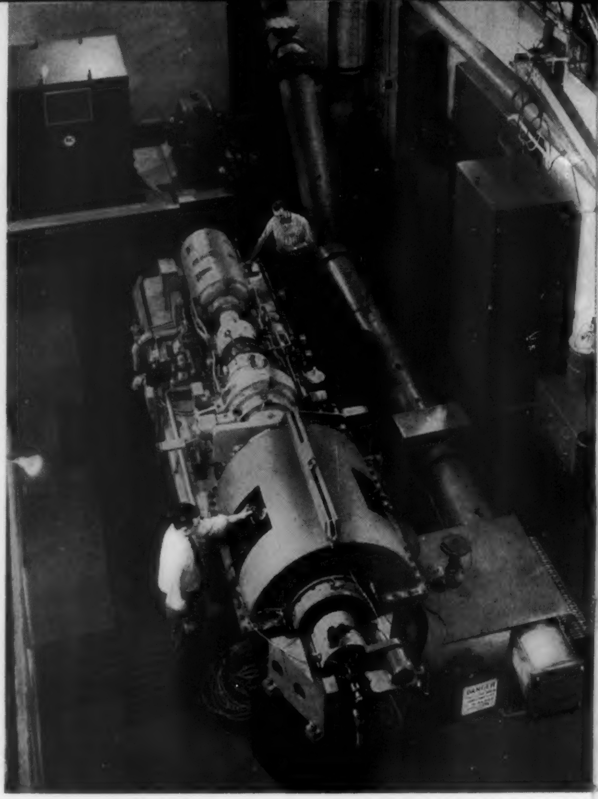
If an operator knows it will be a period of time before the tanks will again need to be charged, a substantial horsepower saving can be obtained by simply closing the suction throttle valve. Because of the limit switch, this will again cause the blowoff valve to lock into the open position and bring the system down to low pressure idling.

Simplicity, high performance and ease of maintenance are therefore the bases of the acceptance of this arrangement. No longer will Convair be required to "wait their turn" in order to get their essential information. The future of this type of tunnel is exceedingly promising and will definitely shorten the time required to produce the missiles and planes that are so necessary for our national defense and the maintenance of our air superiority.

NEW Experimental Laboratory Tests STEAM TURBINE BLADING

by **F. J. ENRIGHT**
Steam Turbine Department
Allis-Chalmers Mfg. Co.

Performance data gained in new experimental laboratory will help steam turbine designers obtain new blade efficiency levels.



TURBINE in foreground operates on air from motor-driven compressor in background. Dynamometer equipment provides stepless speed control for both absorption and motoring.

DESPITE THE HIGH LEVEL of performance attained in the modern steam turbine, the search for improvements cannot be relaxed. There are still many basic problems concerning the fluid mechanics of turbine blading which are not completely understood, and continuous research and development work on blading performance are necessary to achieve even higher turbine efficiencies.

Laboratory serves as a design tool

The new experimental turbine laboratory, shown in Figure 1, was designed to test the type of blading normally used in the high pressure and intermediate pressure regions of a steam turbine. The laboratory's experimental turbine is used for single-stage or multi-stage tests and was built to accommodate both reaction and impulse blading. Provisions were also made for testing the governing stages.

Since the primary purpose of the laboratory is to obtain test results suitable for direct use in turbine design, the flow in the experimental turbine is similar to the actual flow encountered in commercial turbines. Because test results can be determined accurately, the data can be used with confidence. In addition to these technical requirements, considerable thought was given to making the laboratory reliable and easy to operate.

An important consideration was the selection of air as a working fluid for the turbine. Air was chosen over steam for the following reasons:

1. By providing the laboratory with its own air-supply system, it was made independent of other facilities except for a source of electrical power.

2. Because air very nearly obeys the perfect gas laws at atmospheric pressures and moderate temperatures, the calculation of test results is easier than if steam were used.

3. Air does not present the condensation problems that would occur with steam.

Turbine is designed for versatility

Figure 2 is an axial section through the turbine with reaction blading installed. The permanent parts of the turbine are the fabricated outer casing, the front pedestal which supports the casing and the spindle, and the flexible plates which support the casing at the low pressure end. The spindle, the inner cylinder, and the flow guides near the blade path can be changed to accommodate various types of blading. The arrangement for testing impulse blading is shown in Figure 3.

Blade heights and stage diameters were selected so that warped blading normally would not be required. The ratio of blade height to stage diameter is less than 0.15 for all test blading. Because the energy carry-over from stage to stage is relatively large for reaction blading, it was decided that at least four stages would be required for reaction-blade tests. Three stages are used for impulse-blade tests to give approximately the same pressure drops and power outputs that are obtained with reaction blading.

After considering all the test requirements, a turbine with the following characteristics was chosen:

1. The stage base diameter is 19 inches for reaction blading and 24 inches for impulse blading. Blade widths are 0.75 inches, and blade heights can vary from ½ to 3 inches for both types of blading.

2. The turbine operates at a pressure ratio of 2:1, with peak efficiencies occurring near 5000 rpm for reaction blading and 3400 rpm for impulse blading. Maximum speed is 7000 rpm.

3. The power output of the turbine varies from 100 to 600 hp, depending on the type and size of blading being tested.

Considerable effort was devoted to the design of the turbine inlet and discharge passages. Uniform axial flow with a suitable section for instrumentation is required at the inlet to the blading, and axisymmetrical flow with space for instrumentation is required in the discharge passage. Since radial entry to the turbine inlet chamber and radial discharge from the outlet chamber were dictated by mechanical design considerations, circumferential screens are used to distribute the flow evenly.

The flow pattern in the inlet passages was considered important enough to warrant model tests to determine the best screen configuration. Twenty-one model configurations were tested, and the best arrangement was found to be that shown in Figure 4. Two circumferential screens made of perforated sheet steel with 50 percent open area are concentric with the turbine center line. The turbine casing is not concentric with the screens, but is offset so that the minimum flow area around the outer screen occurs opposite the air inlet.

The construction of the experimental turbine is similar to that of commercial steam turbines in some respects. The blades are made of 13-chrome steel, inserted in standard tee slots in the spindles and cylinders. Disks, blade drums, and balance pistons are shrunk on the shafts so that they can be removed if necessary. However, frequent removal is not contemplated because four complete spindles are available and can be rebladed without removing the disks or drums.

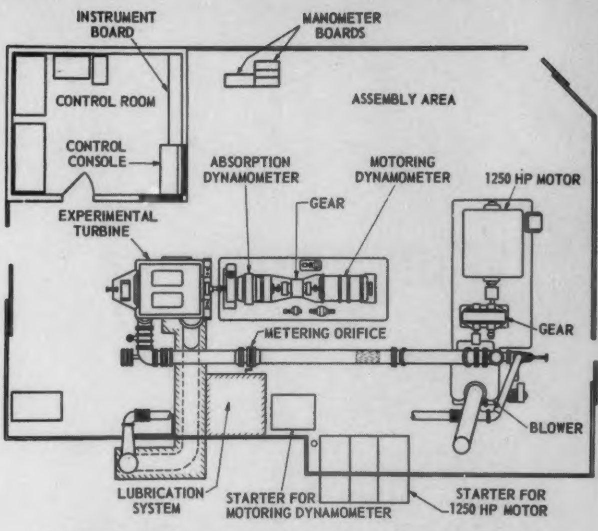
Air leakage from the turbine at the high pressure spindle seal is prevented by introducing sealing air at an intermediate point of the seal. The pressure at this point is made equal to the pressure in the turbine inlet chamber, thus reducing leakage of working fluid to zero.

The air supply to the turbine is obtained from a three-stage centrifugal compressor equipped with adjustable, inlet guide vanes. The compressor is driven at 8500 rpm through a speed-increasing gear by a 1250-hp induction motor.

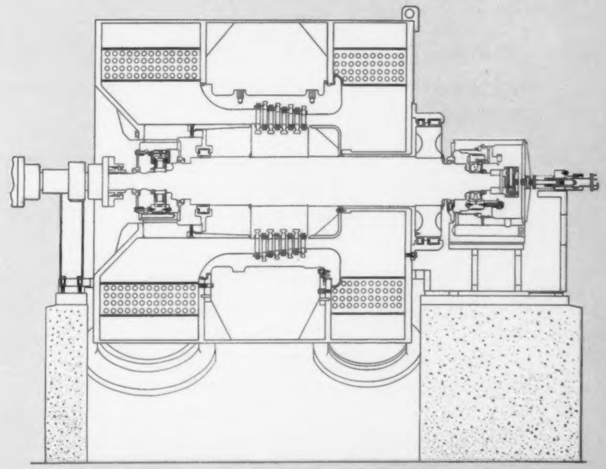
Dynamometer equipment provides test load

The dynamometer equipment consists of three components: an absorption unit, a motoring unit, and a speed-increasing gear. The absorption dynamometer operates on the eddy-current principle with heat being removed by direct cooling of the rotor with water. This unit will absorb 700 hp and has a maximum speed of 7500 rpm. The entire useful speed range of the turbine can be covered by varying the excitation on the field of this dynamometer. Since the absorption dynamometer will not absorb the turbine output at very low speeds, a pin that locks the dynamometer rotor to the stator is provided so that turbine torque can be measured at zero speed.

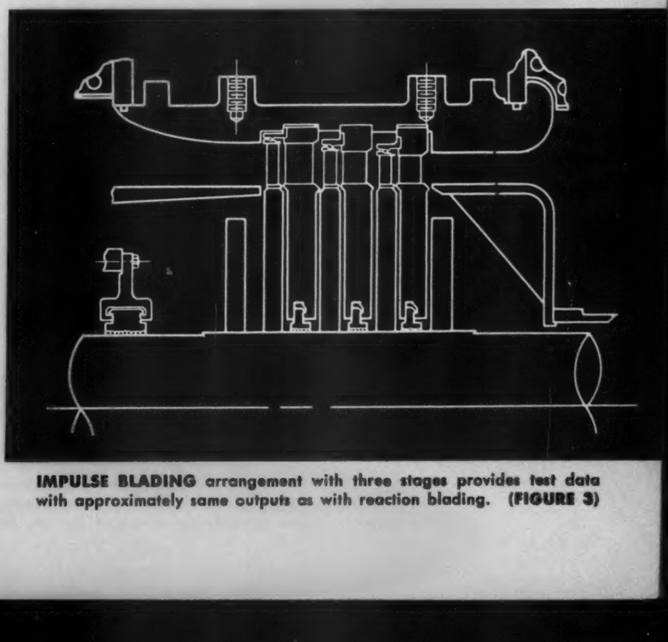
The motoring dynamometer consists of a 100-hp, 3500-rpm induction motor with an eddy-current coupling to regulate speed. A speed-increasing gear located between the motoring and absorption units is used to increase the motoring speed to 6000 rpm.



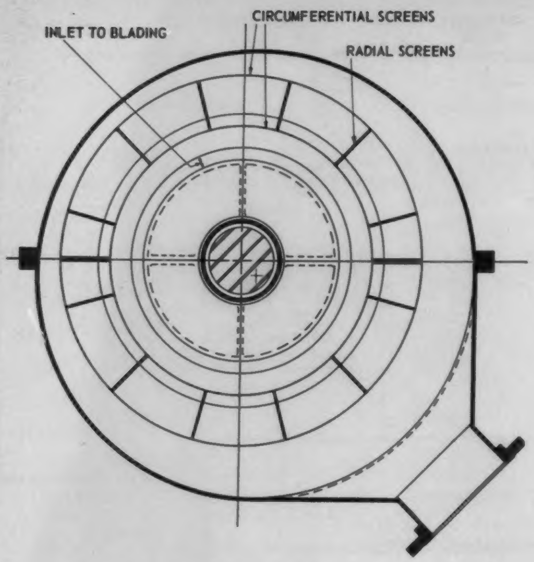
LABORATORY LAYOUT was geared to accessibility of equipment, convenience in getting data and close control by operator in the control room. Arrangement provides sufficient work space. (FIG. 1)



REACTION BLADING is set up in model turbine to provide actual performance data for use in design of these blades. (FIGURE 2)



IMPULSE BLADING arrangement with three stages provides test data with approximately same outputs as with reaction blading. (FIGURE 3)



SCREEN CONFIGURATION is the result of extensive research to perfect the flow pattern of air into turbine blading. **(FIGURE 4)**

Turbine speed is regulated by the dynamometer for both absorption and motoring tests. Test-point speeds can be held within an accuracy of 0.1 percent of the maximum speed of the dynamometer.

Control and instrumentation minimize test time

The entire laboratory can be operated from the control room. Figure 5 shows the operating controls on the console at the right.

In addition to the usual starting and stopping controls for the various laboratory components, a complete set of safety devices is incorporated in the system. Relays are arranged to trip the turbine stop valve and to stop the compressor and dynamometer motors for such faults as overspeed, low oil or water pressure, high oil or water temperatures, high bearing temperatures, low air pressure on auxiliary devices, and low oil levels. Besides shutting down the equipment, the relays operate an alarm system which identifies the fault.

All indicating instruments except manometer boards are located on the instrument panel shown at the left in Figure 5. Data are recorded by hand except for manometer readings. The back-lighted manometer boards are photographed with a 35-mm camera which is operated remotely from the control room.

Sixty-inch U-tube and well-type manometers are used with water or mercury to indicate all test pressures except the differential across the flow-metering orifice. A precision dial manometer with a range of zero to 120 inches of water is used to measure this differential. Precision dial manometers are also used in the control room to establish operating conditions, but are not used to obtain test data.

Iron-constantan thermocouples are used with a 30-point, self-balancing potentiometer for temperature measurements.

Air flow to the turbine is measured with orifice plates installed in the pipe between the compressor discharge and the turbine inlet. The orifice plates are held in a quick-change fixture so that orifices can be changed without disassembling the piping.

Power measured by two methods

Two systems are used to measure turbine power output. For both methods, speed is indicated by an electronic counter which obtains its signal from a magnetic pick-up located near the periphery of a 60-tooth gear mounted on the dynamometer shaft.

The first system of power measurement uses a strain-gage torque meter between the turbine and dynamometer to measure turbine torque. Torque shafts with capacities of 5,000, 12,000, and 30,000 in-lb are available to cover the range of turbine torques from the smallest to the largest blade paths tested. Since the torque-meter brushes are mounted on a lathe crosshead, they can be centered on the torque-shaft slip rings when the turbine spindle is moved axially.

The torque shafts are calibrated with accurate weights. Calibrations were made for both clockwise and counterclockwise torques. The results have indicated that torques can be measured within an accuracy of ± 0.1 percent with careful calibration.

The second method of determining torque uses a hydraulic load cell to measure the force exerted by the dynamometer torque arm. The load cell also is calibrated with weights, and torques can be determined with an accuracy of ± 0.1 percent using calibration curves.

Laboratory is proving valuable

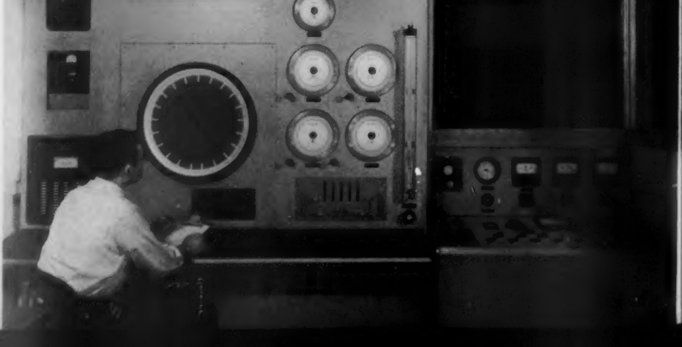
The new experimental turbine laboratory is a permanent installation with a continuous test program. In addition to obtaining performance data on current blade designs, the laboratory is used in conjunction with other test facilities and with theoretical analysis in the development of new blade designs.

The development of the experimental turbine laboratory described in this article was costly in terms of both engineering time and money. The investment in such a facility is justified, however, when it is realized that improvements obtained from laboratory tests will be incorporated in the design of new steam turbines to continue the trend toward more efficient generation of power.

REFERENCE

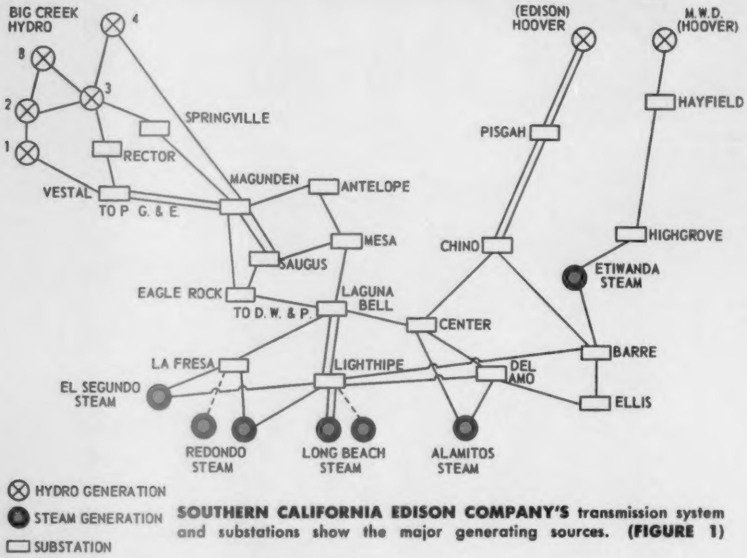
"The Development of an Experimental Laboratory for Performance Tests of Steam-Turbine Blading" (1957 ASME paper), F. J. Enright, available from Allis-Chalmers as Publication A-C 58-9.

TURBINE TESTS are made under surveillance of engineer in control room. Operator has necessary instrumentation and control devices conveniently arranged for his use.



Control equipment is a link in the reliable performance of precision machine tools. Here the dc and regulator panels of a package drive for a large complex milling machine are adjusted and tested before they go into service.
Allis-Chalmers Staff Photo by Harold Shrode

by **E. E. TUGBY**
Substation Engineer
Southern California Edison Co.
and
T. G. A. SILLERS
Chief Engineer
Switchgear Department
Allis-Chalmers Mfg. Co.



SOUTHERN CALIFORNIA EDISON COMPANY'S transmission system and substations show the major generating sources. (FIGURE 1)

FACTORY AND FIELD

High Voltage Short-Circuit Testing

Field testing by Southern California Edison Company supplements product development laboratory efforts.

CIRCUIT BREAKER DEVELOPMENT requires highly repetitive testing under controlled conditions to establish performance capability that will meet system requirements. Testing operations can well be provided both in factory and field laboratories to satisfy manufacturers and users that resultant products are adequate and that service conditions are duplicated.

Factory short-circuit test facilities are usually machines and equipment of special design, suitably isolated so that operating failures do not interfere with any form of power supply. The frequency and time of test operation are limited only by equipment performance and the designers' skill and ability. Special test circuits and special circuit modifications can be used without endangering normal operating circuits. Economical considerations limit the test capacity which can be made available.

Tests conducted both in field and laboratory

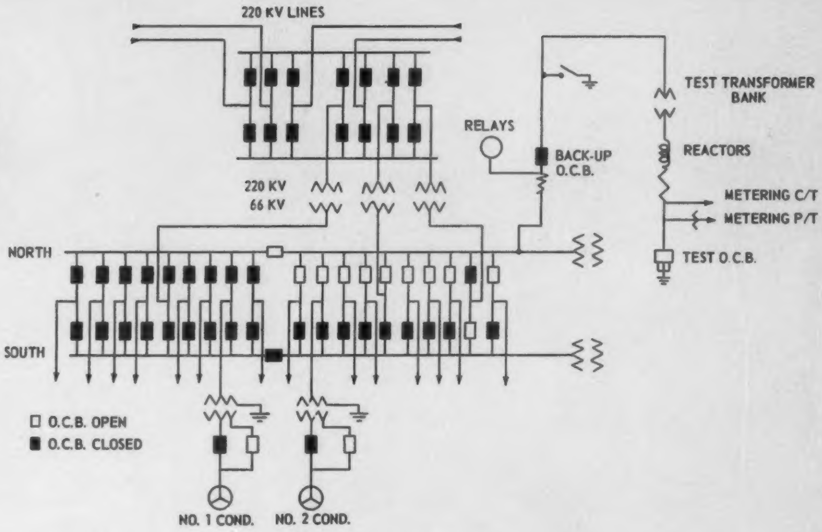
Field test facilities normally include standard machines and circuits as used for power generation and distribution. Testing schedules must consider power production, and the time and frequency of test operations must conform to system load requirements. Instrumentation for field tests may be more limited than provided for factory laboratory tests. Factory laboratories are best suited for the development type of testing. Field tests are most desirable for proof-testing. Capacity limitations are normally based on system capacity and the amount of short-circuit capacity that the utility cares to make available.

Certain operations such as line dropping, switching unloaded cables and load switching are not easily duplicated in laboratories. These operations involve little system hazard and require the minimum of special field setup, and consequently are desirable field tests. The performance of system equipment other than the circuit breaker is of particular interest in these types of test operations.

Field short-circuit operations subject the system to conditions encountered when faults occur. The performance of the protective system is checked and correct operation assured during actual faults.

Utilities have had varying policies with regard to system short-circuit testing, and at various times field testing facilities have been made available with comprehensive control and instrumentation, and with output much in excess of laboratory capacity. These facilities have contributed immeasurably to the development of improved switchgear. The Southern California Edison Company has field testing installations which have been in operation

AVAILABLE THREE-PHASE fault duty at the 220-kv bus at Center Substation is 6340 mva. Net 66-kv three-phase short-circuit capacity is 1225 mva, with two 90-mva transformer banks isolated for test service. When tests are in progress, the double 66-kv bus permits load to be carried on one bus while test circuit is carried on the other bus to minimize system voltage dip. **(FIGURE 2)**



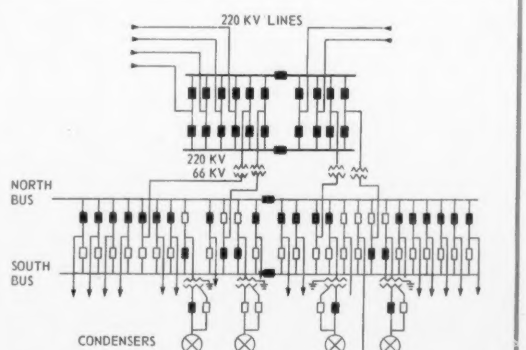
since 1954. One was established to test breakers for 4-kv and 12-kv service to capacities of 180 mva, three phase and 250 mva, three phase, respectively. To minimize system disturbance, all testing is performed with an isolated bus from the 220-kv transmission system used as a source. Center Substation was selected for the lower voltage test installation, because transformer capacity can be made available to the test station during light load periods. Figures 2 and 3 are diagrams of two substations at which short circuit tests are regularly performed.

Test facilities are versatile

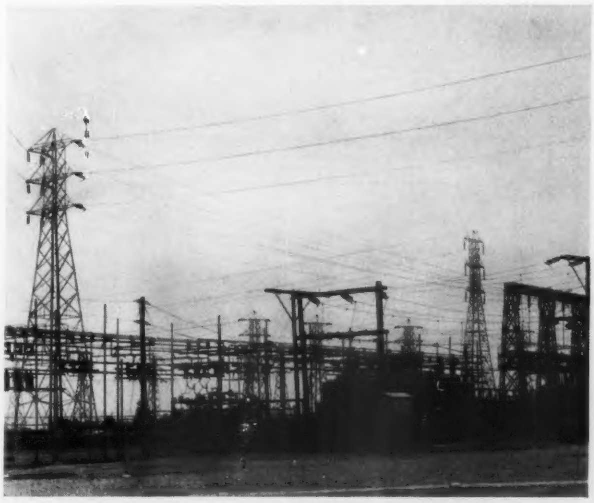
Southern California Edison Company operates a 220-kv transmission system with 2,208,741 kva of steam and diesel generation and 1,021,650 kva of hydro generation, connected as of November 1, 1957. At Center Substation, shown in Figure 4, two 90/120-mva transformer banks, 220 kv to 66 kv, can be made available for test supply, with 1225-mva short-circuit capacity available on the 66-kv bus. Backup protection is provided by an oil circuit breaker in the test rack.

Various transformer combinations have been used for power supply at service voltages. Banks ranging from 6000 to 18,000 kva for 4-kv tests and 30,000 and 60,000 kva for 12-kv tests have been used in various combinations.

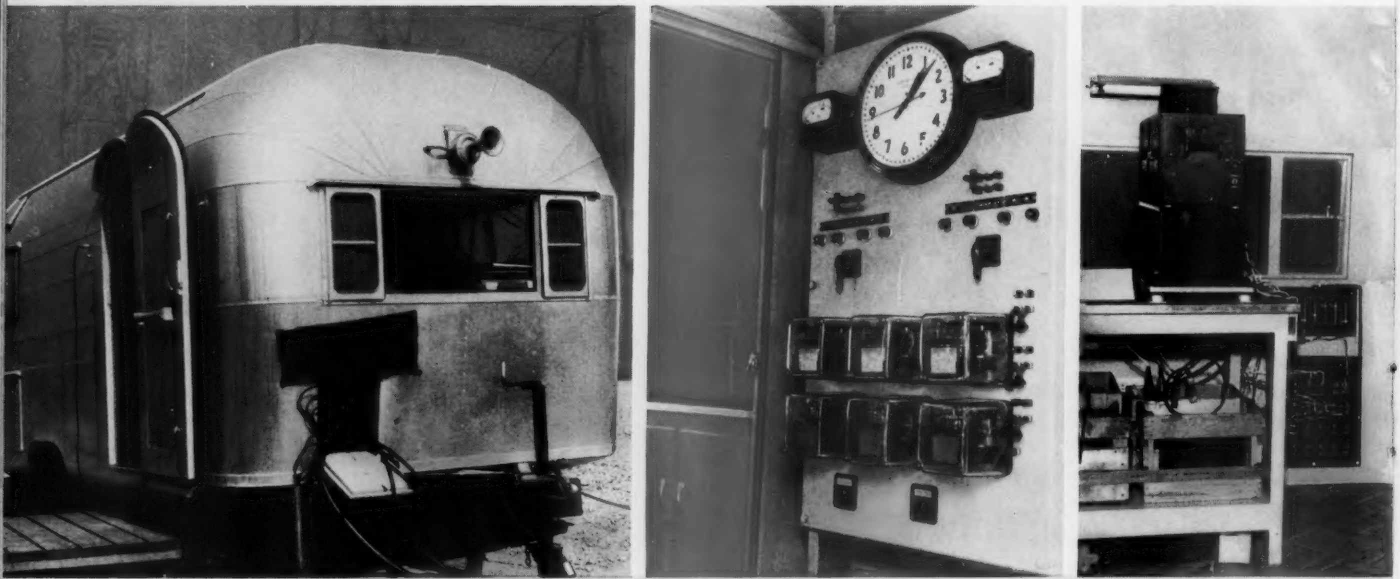
Standard design transformers have been used for the test service, providing confirmation of the transformer per-



THE MAJOR STEPS of test current at the Laguna Bell Substation are secured by connecting combinations of 220/66-kv transformers and condenser transformer banks to isolated bus. Further current adjustment is made by combinations of reactors in the ground lead. It is possible to secure test faults from 295 mva equivalent three phase to 2850 mva equivalent three phase as a maximum. **(FIG. 3)**



THE GENERAL TEST AREA AT THE CENTER SUBSTATION contains four test bays on the left, three 10,000-kva, 66/12/4-kv transformers in the center, a 66-kv backup oil circuit breaker to the right and 66-kv double bus in background. **(FIGURE 4)**



THE GENERAL AREA in the Mobile Test Station trailer contains oscillographic equipment in the front, control panel and desk space in the center, and film processor and darkroom in the rear. **(FIGURE 5)**

formance. Before these transformers are placed in commercial service, they are untanked and inspected for indications of magnetic stresses. Insulators, disconnects, potential and current transformers are standard equipment except for three high-ratio current transformers.

Mobile station carries test equipment

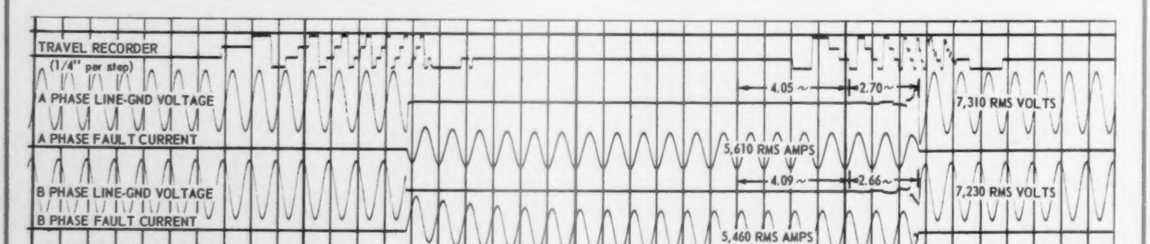
A mobile test station, shown in Figure 5, contains control and recording equipment for all tests. A standard two-wheeled, 18-ft house trailer, equipped with oscillographs, control panel and darkroom equipment, has proved to be versatile and efficient in this test work. A 15-element oscillograph with daylight loading film magazines provides the oscillographic records of tests.

The control panel is provided with control switches for backup and test breakers. Overcurrent relays with instantaneous trips are provided for the test circuit breaker. Low energy relays without instantaneous trips are provided for the backup breakers.

An "oscillograph processor," capable of processing paper negative strip at speeds of from two to ten feet per minute, makes records available for examination within a few minutes of test completion. Figure 6 is an oscillographic test record. Travel recorders of the four-step type, readily adaptable to any circuit breaker, are proving to be very successful.

Test facilities increased

Room is provided for transformers of various ratings. Three single-phase tapped reactors plus variations in transformers and transformer connections provide current variations as desired. Four test bays for 4 kv and 12 kv are



A TYPICAL TEST RECORD, made on the oscillograph processor, shows an "open-close open" interrupting test on a 12-kv oil circuit breaker, with a 199.4-mva, three-phase equivalent fault. **(FIGURE 6)**

provided, each equipped with isolating disconnects. Power supply is from the 66-kv system through a 66-kv backup breaker to a pipe rack bus structure.

Initial test values of 180 mva at 4 kv and 250 mva at 12 kv have now been increased to 313 mva by installation of a 30,000-kva, 66/4/12-kv transformer bank as a permanent unit. Additional 12-kv capacity to 515 mva is available by paralleling this test circuit with a 60,000-kva synchronous condenser bank.

At 69 kv, more desirable test arrangements were available at the Laguna Bell Substation than at Center Substation, and these operations were moved to Laguna Bell Substation. Figure 7 is the Laguna Bell test area. Four 90,000-kva, 220/69-kv transformer banks and four synchronous condensers are in service at this station, and 220/69-kv transformer banks and associated bus may be isolated for testing during off-peak periods. Test capacity up to 2850 mva, three phase equivalent is available at the test bus. All testing at this voltage has been done on a single phase to ground basis.

Current control is obtained by variations in the transformer and condenser connections and by reactors in the ground load. A synchronous timer for controlled tripping of the test breakers is available.

Tests are carefully programmed and pertinent test and performance records are made as shown in Figure 8. Representatives of manufacturers whose equipment is under test are always invited to witness the acceptance tests on their equipment and exchange technical data with the test engineers. Operations conform to the standard operating practices of the Southern California Edison Company.

Additional test programs are planned on new circuit interrupting devices as they become available. Performance of equipment other than circuit breakers is also determined during these test operations.

THE GENERAL TEST AREA at the Laguna Bell Substation has an oil circuit breaker test structure with reactors in the foreground and a station double bus in the background. **(FIGURE 7)**

The Southern California Edison Company feels that these testing programs provide an effective proving ground for circuit breakers and interrupting devices. Equipment with an adequate reserve of capacity establishes confidence that it will meet its full rating in times of system trouble. Assistance is provided in the development of new devices adequate for modern high quality electrical service.*

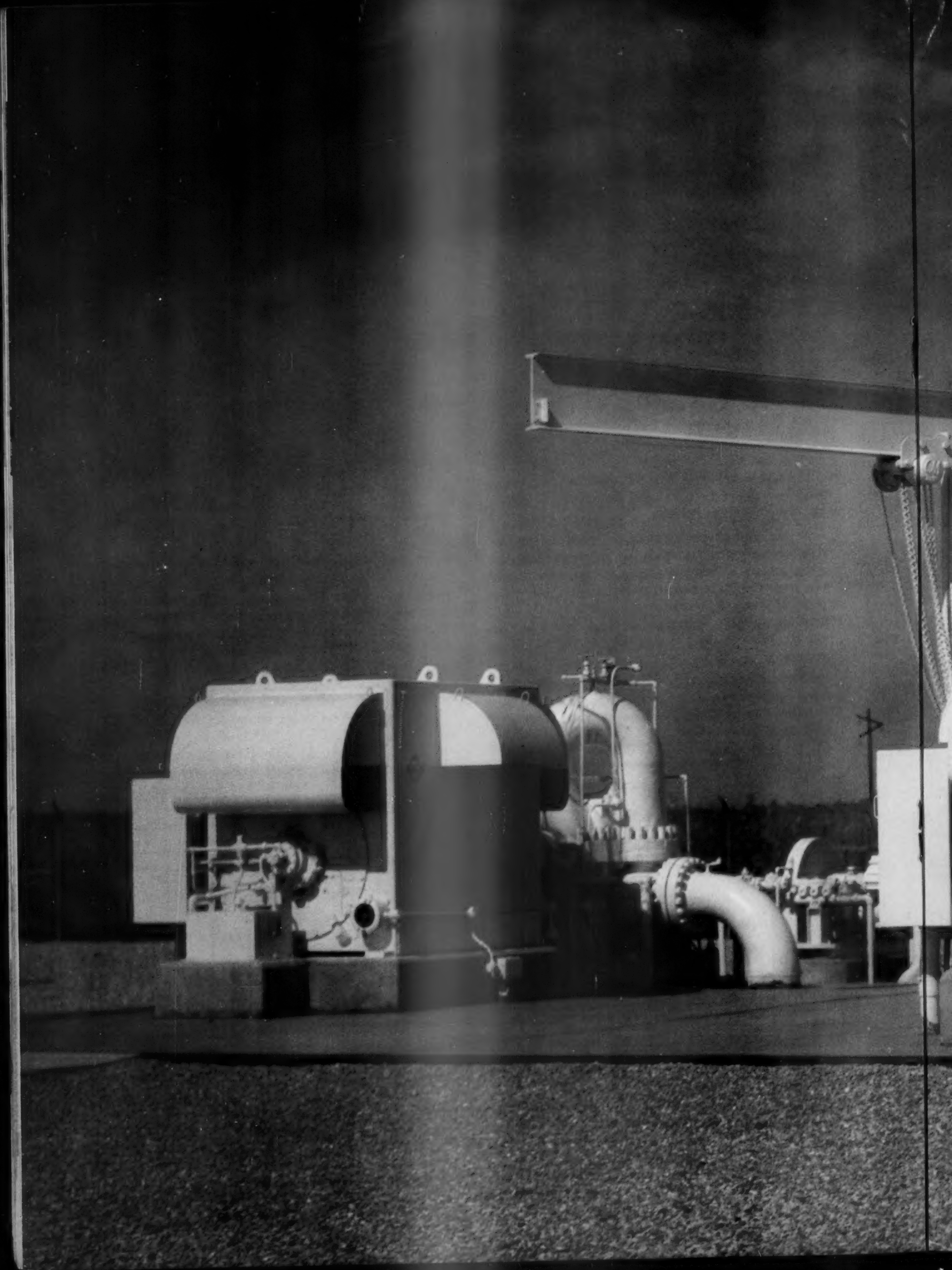
* "High Voltage-High Capacity Interrupting Test Facilities of Southern California Edison Company" by E. E. Tugby, AIEE Paper 57-916.

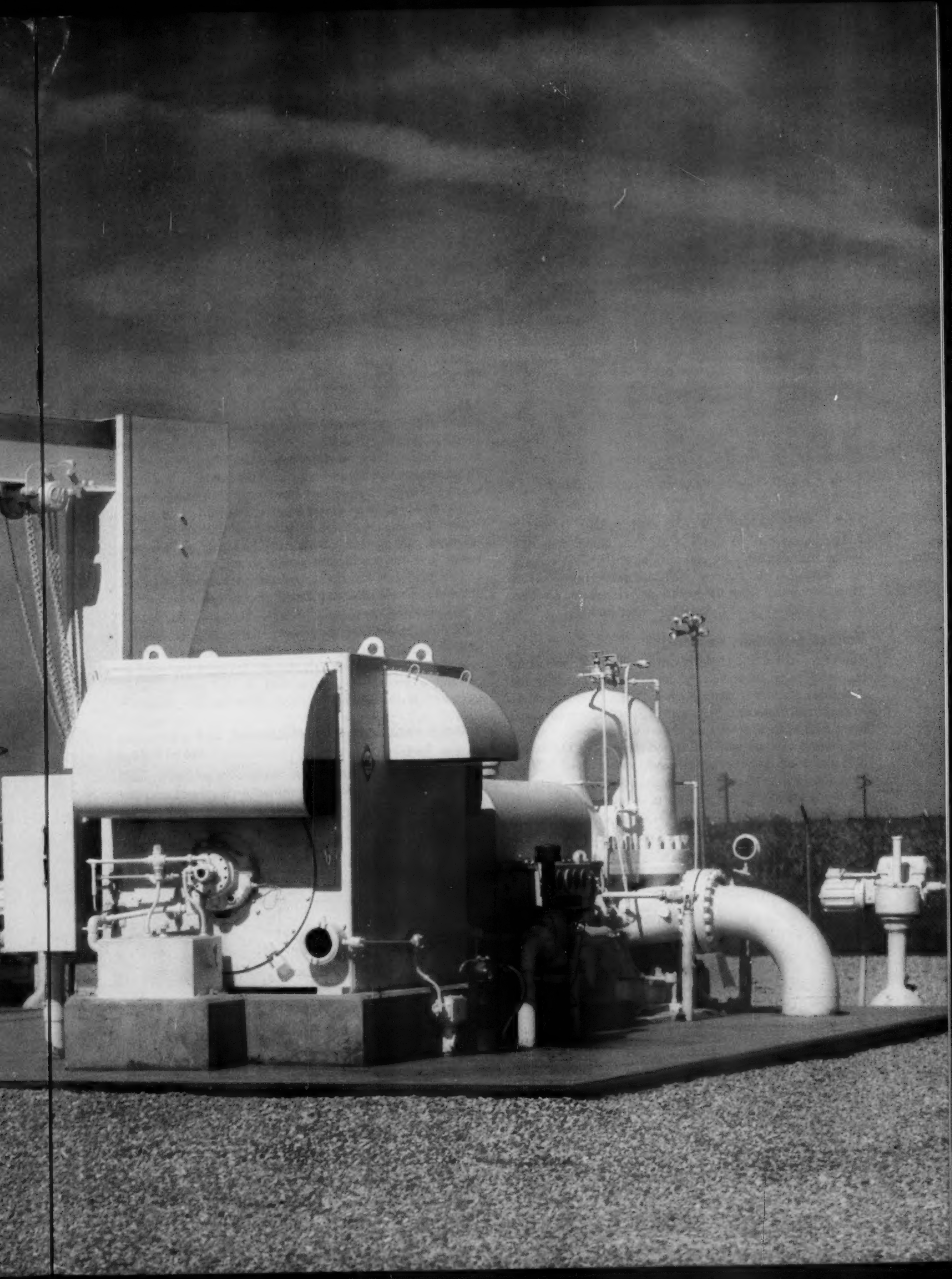
12 KV TEST ANALYSIS — NO. 2

Group	Reqd. MVA	Reqd. %	Transformer Bank	% Bank	% Syst.	Reactor %	Reactor Ohms	Total % Calc.	Total % Act.	MVA Calc.	MVA Act.	Amps Ø Calc.	Amps Ø Act.	Amps Ø to Neutral Calc.	Amps Ø to Neutral Act.	Equivalent 3 Ø MVA Calc.	Equivalent 3 Ø MVA Act.
A	17	2540	1500 kva 3-500	2300	76	610	1.76	2986	3060	16.7	16.3	786	765	792	740	16.9	15.8
B	20	2500	1500 kva	2300	76	0	0	2376	2480	21.1	20.2	987	945	998		21.2	
C	62	800	30,000 kva 3-10,000	117	76	610	1.76	803	865	62.3	58.0	2920	2710	3050	2870	64.3	61.2
D	96	520	30,000 kva	117	76	337	.97	530	520	94.4	96.3	4440	4510	4640	4730	99.0	99.5
E	125	400	30,000 kva	117	76	229	.66	422		118.5		5570		5900		126.0	
F	150	333	30,000 kva	117	76	161	.465	354	350	141.0	143.0	6640	6700	7140	7200	152.0	152.0
G	175	285	30,000 kva	117	76	98	.281	291		171.5		8070		8830	8700	188.0	183.0
H	215	235	30,000 kva	117	76	41	.118	234		214.0		10,100		11,200	10,900	239.0	231.0
J	250	200	30,000 kva	117	76	0	0	193		259.0		12,200		13,950	12,600	297.0	268.0

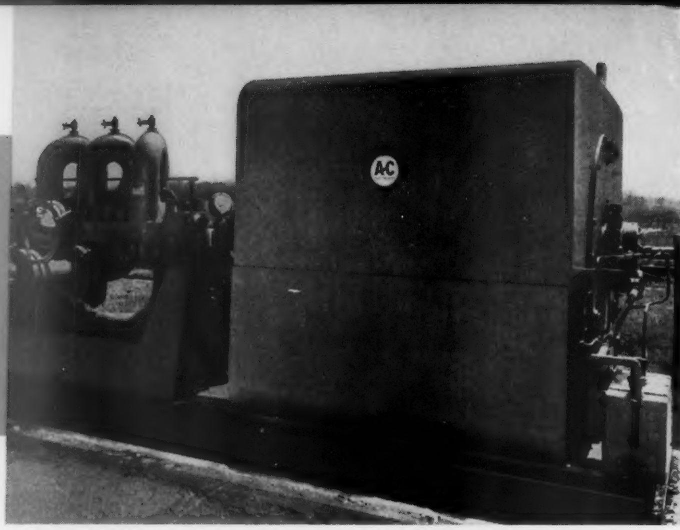
TEST ANALYSIS contains equipment performance record. **(FIGURE 8)**

NOBODY HERE — Pumping station is completely automatic; controlled by microwave system. Two 2500-hp, 4160-volt, 1785-rpm squirrel cage induction motors drive crude oil pipe line pumps for The Ohio Oil Company. Motors are unique in that standard drip-proof enclosure is modified by addition of sheet metal hoods over air intake and exhaust openings. Coils are *Silco-Flex* insulated for fully reliable outdoor service.
Allis-Chalmers Staff Photo by J. E. Gosseck

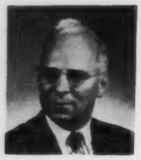




Operating Induction Motors on UNDER and OVERVOLTAGE



LONGER SERVICE LIFE and top efficiency can be expected when the motor's voltage is held at its nameplate value. Unattended, this 1000 hp, weather-protected pipeline motor operates on 2300 volts.



by **R. C. MOORE**
Motor and Generator Dept.
Allis-Chalmers Mfg. Co.

Motor terminal voltage may vary with changes in loading on your distribution system. Here is what can happen.

THE IMPORTANCE OF SYSTEM voltage control is seldom thought of in terms of efficient motor operation. Motor characteristics are seriously affected when the system voltage differs from the nameplate rated values. Most important, from an operational viewpoint, are electrical, mechanical and thermal characteristics.

Voltage deviations analyzed

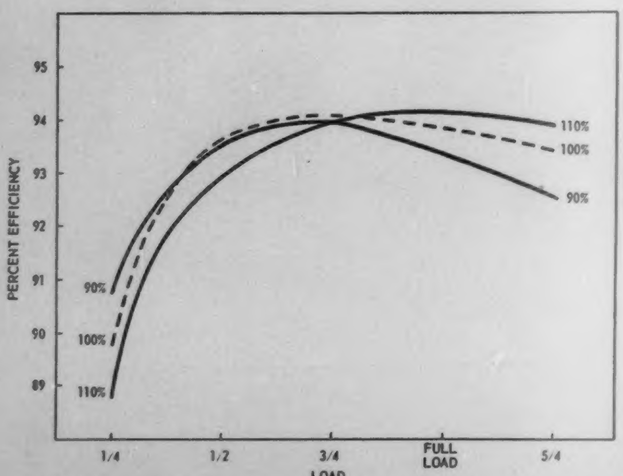
The effect of voltage changes can be shown by analyzing the usual permissible voltage deviations outlined by the American Standards Association.[1] These standards state in effect that induction motors shall operate successfully at rated load with a voltage variation up to plus or minus ten percent of rated voltage at rated frequency. The standards also state that the performance within these voltage variations will not necessarily be in accordance with the standards established for operation at rated voltage.

From an electrical viewpoint, interest centers on the effects of under and overvoltage operation on motor efficiency and power factor. From a mechanical aspect, the motor speed and torques are of concern. These torques are the starting, pull-up, breakdown, and load running torques. In addition, noise or vibrations may develop as a result of voltage changes. Thermal problems may also arise because of increased losses or changes in the distribution of these losses. Impaired motor ventilation resulting from a drop in speed may also add to thermal problems. When the motor load current increases with voltage change, added coil copper losses will cause accelerated insulation deterioration of old-style motors and will reduce their service life.

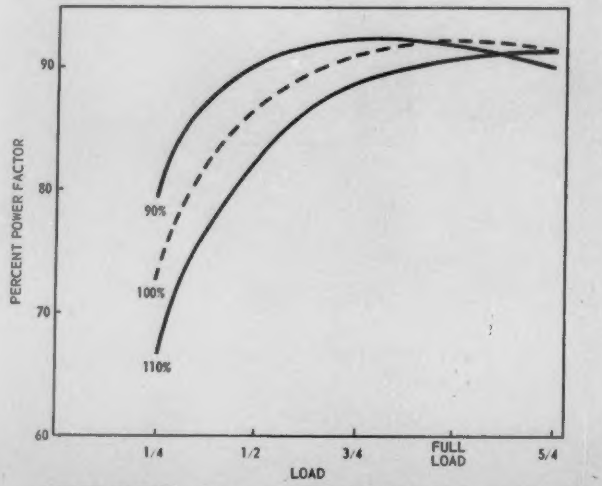
Several examples using a 40-degree rise motor for 2300-volt, three-phase, 60-cycle operation illustrate the effects of voltage variation on machine characteristics. In each case a breakdown torque value in the range of 230 to 250 percent, locked-rotor currents 600 to 650 percent, and locked-rotor torques in the vicinity of 100 percent are used.

Voltage variations alter efficiencies and power factors

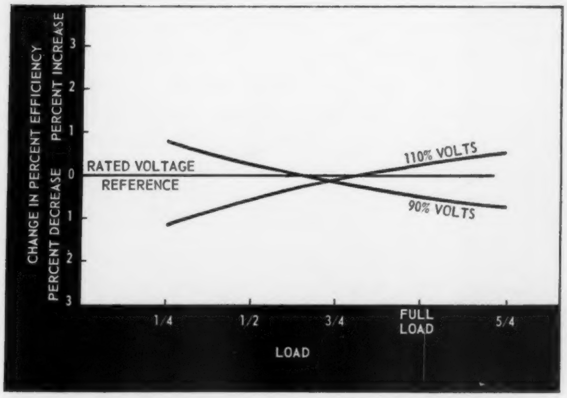
A trend in efficiency and power factor with plus or minus ten percent voltage change is shown in Figures 1 and 2. It is interesting to note that when the voltage is reduced, the maximum values of both efficiency and power factor are shifted toward the lower horsepower capability range.



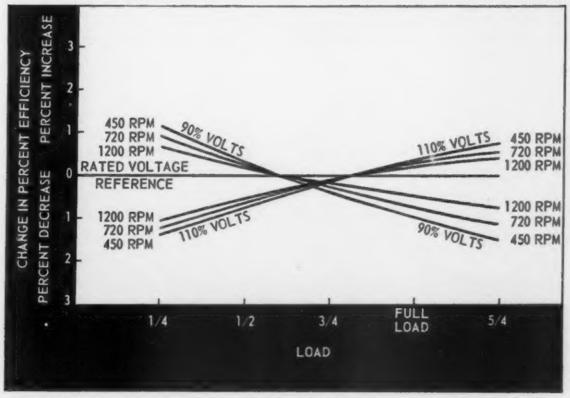
EFFICIENCY at 90, 100, 110 percent voltage is given for 600-hp, 1200-rpm cage motor. Droop is greater at reduced voltage. **(FIGURE 1)**



POWER FACTOR with overvoltage and light load is lower than with undervoltage as shown for 600-hp, 1200-rpm cage motor. **(FIGURE 2)**



EFFICIENCY CHANGE for undervoltage and overvoltage in motor of Figure 1 is compared to rated voltage for the motor. **(FIGURE 3)**



DIFFERENT SPEED RATINGS (synchronous speeds shown) for a 600-hp, 60-cycle motor have different efficiency curves. **(FIGURE 5)**

Similarly when the voltage is raised, the maximum values are shifted toward the higher horsepower range.

The figures also show that the power factor in an underloaded motor operated at higher than rated voltage will drop appreciably. Furthermore, the efficiency will fall in the same manner, but not in the same amount. In Figure 1 the motor efficiencies at 75 percent load are almost the same at rated, 90 percent and 110 percent voltage. On overvoltage the increased iron losses resulting from higher magnetic densities are offset by lower copper losses. On the other hand, at lower voltage the reduced iron losses are compensated for by the increased copper losses.

Data from Figures 1 and 2 are given in slightly different form in Figures 3 and 4 to show the change in efficiency and power factor for operation at the different voltage.

Machine speeds and sizes considered

Changes in efficiency and power factor for plus or minus ten percent voltage variation are shown in the calculated curves for the two cases.

The first case is shown in curves in Figures 5 and 6. Changes in efficiency and power factor with plus or minus ten percent voltage variation are shown for several cage-type induction motors having the same horsepower but different synchronous speeds.

For the voltage variations assumed, the change in percent efficiency is not so great as the change in percent power factor. As previously explained in the case of efficiency, the change in magnetic (iron) losses when the terminal voltage departs from the nameplate value is offset by a change in copper losses. This characteristic can be illustrated in part to show the copper loss change by calculating the change in full-load current. For the 600-horsepower, 1200-rpm motor of Figures 1 and 2 the full-load current is calculated as follows:

Full-load amps at 100 percent volts =

$$\frac{600 \times 746}{\sqrt{3} \times 2300 \times .938 \times .916} = 131$$

where .938 and .916 are efficiency and power factor, respectively, from Figures 1 and 2.

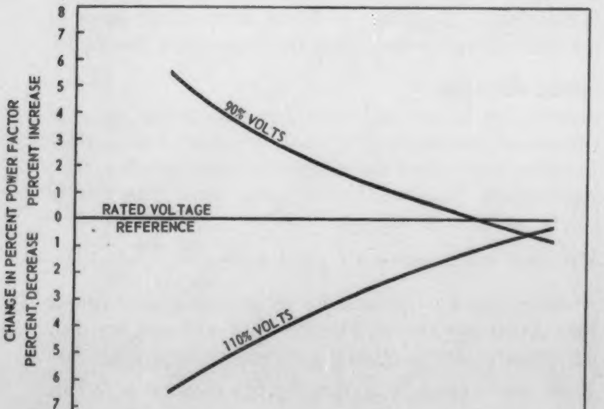
Likewise:

Full-load amps at 90 percent volts =

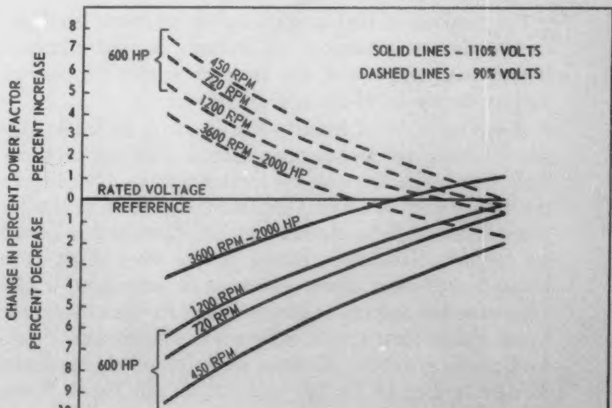
$$\frac{600 \times 746}{\sqrt{3} \times .90 \times 2300 \times .934 \times .910} = 147$$

Full-load amps at 110 percent volts =

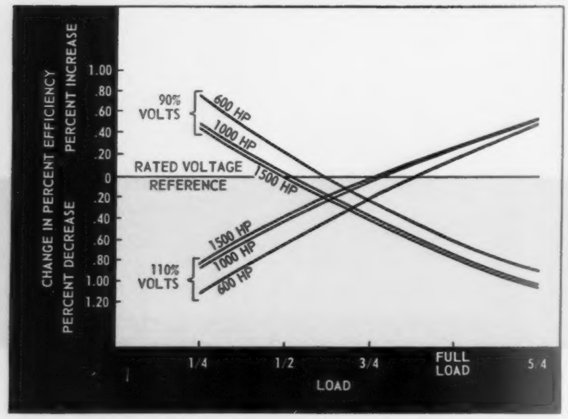
$$\frac{600 \times 746}{\sqrt{3} \times 1.10 \times 2300 \times .942 \times .902} = 120$$



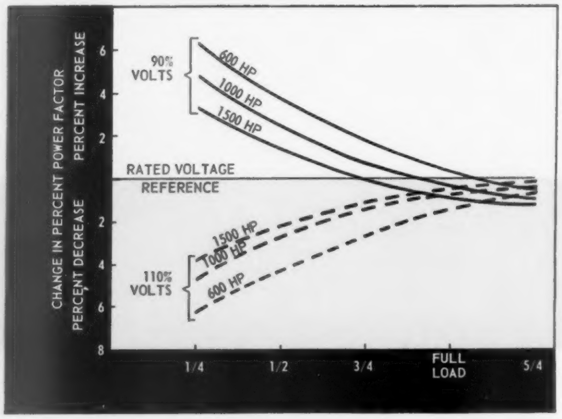
POWER-FACTOR CHANGE with voltage change in motor of Figure 2 can be shown clearly when referred to rated voltage. **(FIGURE 4)**



POWER-FACTOR CURVES for motors with different speed ratings differ for undervoltage and overvoltage conditions. **(FIGURE 6)**



EFFICIENCY CHANGES for various size motors vary differently when supplied with undervoltage or overvoltage. **(FIGURE 7)**



POWER-FACTOR CHANGES for various size motors vary differently when supplied with undervoltage or overvoltage. **(FIGURE 8)**

The result of overvoltage on motor power factor is an increase in motor magnetic densities both in the air gap and in magnetic saturation. Magnetizing current is thereby increased. Conversely, an undervoltage condition results in a lower magnetizing current. Besides a change in magnetizing current, a change in circle diameter[2] also occurs to influence changes in power factor.

The second case is illustrated in Figures 7 and 8. Variations in efficiency and power factor are shown for motors having the same synchronous speeds but varying in size, that is, in horsepower ratings. The general interpretation of the curves is the same as for the previous case.

Voltage changes change torques

When starting a motor on off-normal voltages, the lock-rotor, pull-up and breakdown torques must be considered. Standards[1] do not set voltage variation limits. Voltage drop at the motor terminals is determined by motor current together with such factors as system size, line drops and transformer impedance. During the starting-up period the motor current continuously drops, so that the voltage drops in associated equipment, such as lines, transformers, are gradually lessened. Voltage regulator influence also enters the problem. The larger horsepower motors will produce the most noticeable drops in voltage. As a motor speeds up, the motor terminal voltage will rise and no single value of motor voltage will apply from standstill up through the accelerating range.

For purposes of analyzing the induction motor itself, excluding lines, transformers, and so forth, an approximation is frequently made with the assumption that the torques vary as the square of the applied voltage.

A typical curve of locked-rotor torque is shown in Figure 9. Since the locked-rotor current does not vary linearly, that is directly, with the applied voltage, a bending in the voltage-current curve is apparent. The bending is more pronounced in some motors than in others and is caused by magnetic saturation, mostly in the rotor teeth. The actual locked-rotor current, because of saturation, is 600 percent at full voltage. Corresponding to this current, the actual locked-rotor torque is shown as 100 percent of full-load running torque. Torques are infrequently expressed as a percentage of the full-load torque. In Figure 9 two torque curves are shown. If the motor current had dropped in direct proportion to the applied reduced voltage as shown in the dashed voltage-current curve, the developed torque would be expected to drop as the voltage squared. However, the current drops off faster than the voltage, as shown by the solid voltage-current curve. Many engineers, therefore, prefer to calculate the torque reduction as the square of the current, since the current which flows through the motor is more nearly related to torque.

Pull-up and breakdown torques are altered

A complete speed-torque curve of a cage-type induction motor is shown in Figure 10, and the principal torques are indicated.[3] In the absence of specific motor design information for a speed-torque curve, the assumption is usually made that the motor torque varies as the square of the applied voltage. Since the motor current and magnetic saturation drop off at the higher motor speeds, the assumption appears to be closer to actuality during acceleration. Motor cage temperature will rise during acceleration of the motor and its connected load, thus causing the torque to vary.[4, 5]

In certain applications, notably in power plants, voltage dips up to as much as 20 percent have been anticipated. Considerable torque allowance in the complete full-voltage speed-torque curve must then be made to provide adequate torque when the voltage dips to the expected values.[6] Starting a motor on undervoltage rather than overvoltage is more likely to produce problems with voltage drops in the line, through transformers or from supply line faults.

Speed changes

A reduction in terminal voltage reduces motor speed in the normal running range. Conversely, there is an increase in motor speed when the terminal voltage exceeds normal rated values. The variation of motor speed with voltage can be analyzed as follows:[7]

$$\text{Kw rotor loss} = \text{motor air gap kw} \times \frac{\%\ \text{slip}}{1 - \%\ \text{slip}}$$

where air gap kw = stator kw input minus stator copper loss. As air gap kw and ($1 - \%$ slip) will vary but little for efficient motors driving a constant torque load, then

$$\text{Rotor loss} = \text{rotor } I^2R = \text{rpm slip} \times \text{a constant} = \%\ \text{slip}.$$

As the rotor current will increase proportionally (approx.) to 1/voltage, then % slip varies as $1/(\text{voltage})^2$. This statement is sometimes found to be a useful approximation to determine speed drop with voltage change.[8]

This derivation is based on a constant-torque load. In applications such as fan loads there will be a reduction in connected load horsepower requirements when the speed drops; therefore the drop in speed for such drives will be somewhat less than the formula indicates.

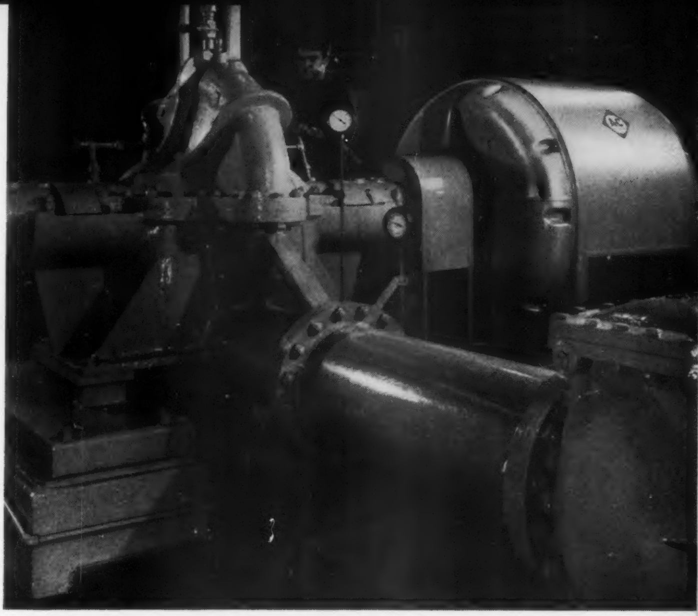
Off-normal voltage influences noise

Some motors may operate with little or no perceptible magnetic noise under normal voltage and load conditions. While in some machines an increase in voltage may cause no noticeable changes in magnetic noise, in others the same change in voltage may produce magnetic noise problems. The difference is, of course, due to increased mechanical vibration or movement caused by increased magnetic forces occurring at overvoltage. Increases in noise will occur with overvoltage, while effects of magnetic noise, if any, are usually lessened with undervoltage.

Insulation life affected

Since motor load currents can increase on undervoltage, the increased internal coil copper temperatures may cause insulation deterioration in old-style motors. While it is difficult to set an exact rate of deterioration, the tendency still exists and will work to the detriment of insulation life, except in the very modern motors having new types of insulations which are not affected by temperature changes of this magnitude. Ordinarily the effect of reduced ventilation resulting from machine speed drop would be expected to be relatively insignificant. Totally-enclosed fan-cooled motors which depend on two sets of fans for ventilation, however, will experience greater than normal changes with speed drop.

Motors are designed with a given voltage rating and deviating from this voltage will produce some loss in service life of the motor, impair its efficiency, adversely affect the system power factor and possibly create noise problems. For these reasons it is wise to check motor terminal voltage to be sure it matches the nameplate rating.



OUTPUT of 10,450-gpm, 180-ft head centrifugal pump can be maintained only if the voltage of its 600-hp, 700-rpm squirrel-cage motor is held at its nameplate value. Reduced voltage will reduce motor speed.

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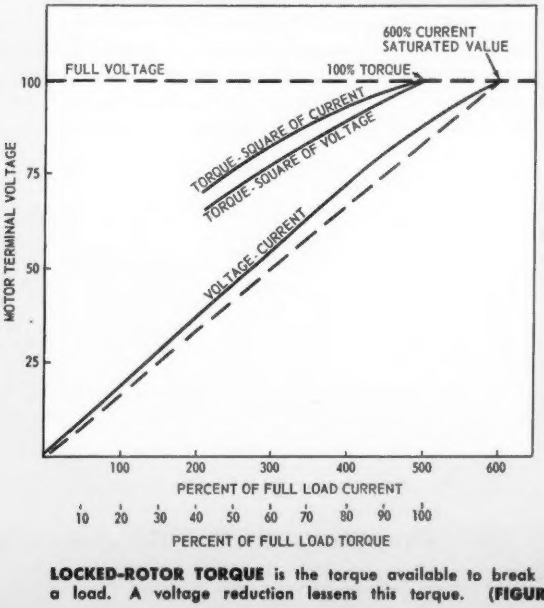
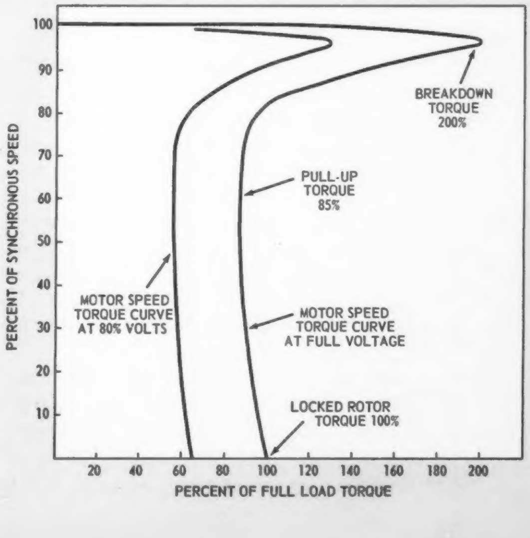


LOCKED-ROTOR TORQUE is the torque available to break away a load. A voltage reduction lessens this torque. **(FIGURE 9)**



VOLTAGE REDUCTION causes a shift of the entire speed-torque curve, thus lowering the available torque. **(FIGURE 10)**

PANCAKE MOTORS

Solve Space Problems

by **V. G. HONSINGER**
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Development Laboratory
Norwood Works
Allis-Chalmers Mfg. Co.

Pancake motors have successfully been applied to limited space applications in the machine tool industry. Several special considerations influence their construction.

ENCLOSED, fan-cooled radial air-gap motor driving milling machine is a typical application. Pancake motors will fit in restricted spaces. **(FIGURE 1)**

THE INCREASING TENDENCY to integrate electric motors with the machines they drive has caused a growing awareness of styling and a continuing demand to create smaller and smaller motors. Since the NEMA flange-type C, D, and P motors were standardized they have become very popular, especially with machine tool and pump builders. The pancake motor, so called because of its flatter shape and larger diameter, has paralleled the growth of the standard NEMA motor.

There are at least three basic forms of induction pancake motors. They are the radial air gap, the inverted radial air gap and the axial air gap, shown in Figures 2, 3 and 4. These basic forms may be given various enclosures, such as open, drip-proof, enclosed-nonventilated, enclosed fan-cooled, and so on.

Use of a pancake motor is justified when a standard motor is too long for mounting on a machine. Performance of this type motor is similar to that of the NEMA standard motor. Efficiencies, power factors, locked-rotor torques, and currents are not greatly affected.

A characteristic common to all pancake motors, resulting from their large diameter, is their large inertia. Consequently, these motors are not generally desirable in reversing service or quick accelerating duty. Pancake motors tend to require less steel but more copper than standard motors, and their large diameter also accentuates the end-extension copper. This part of the winding does not contribute toward torque generation, but is parasitic and a source of I^2R loss.

Computations decide radial air-gap motor size

The radial air-gap pancake motor, shown in Figures 2 and 5, is little more than a conventional motor made flatter. The big problem in the design of pancake motors is how to reduce the length of the copper end extensions. This problem is accentuated in pancake motors because the amount of copper in the end extensions is proportionally larger than in other motors. It is this factor which places a limit on how flat a pancake motor can be built. Consider that a conventional motor is made flatter by design with the motor volume remaining constant. As the motor grows flatter, the active steel grows flatter, but the copper in the end extensions becomes longer. The motor length is finally limited at the point where the increasing copper length in the end extensions overshadows the decreasing core length. The relationship between core length and copper end-extension length and how their combined length passes through a minimum may be demonstrated analytically. Refer to Figure 6 for definition of terms. The resulting equations are of considerable use in proportioning the rotor core and in determining how to secure the flattest possible motor.

For a stationary value of rotor D^2L, call it $\varnothing$, the core length will be $\frac{\varnothing}{D^2} = L$. The end extensions are considered here to be V-shaped, although any shape that is an increasing function of diameter D could be used to prove the theorem. The winding diameter is kD and is slightly larger than the rotor diameter D by approximately 20 percent. From the geometry of Figure 7, the axial length of the end extensions (both ends) is $2\beta + kD \sin\frac{\pi}{P}\,\rho \tan\alpha$.

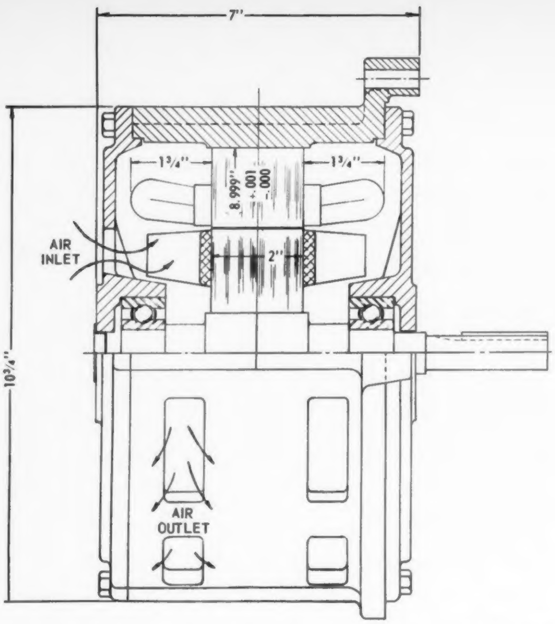
The total length of core and copper winding $\wedge$ is then

$$\wedge = \frac{\varnothing}{D^2} + 2\beta + kD \sin\frac{\pi}{P}\,\rho \tan\alpha \qquad (1)$$

the condition for minimum length is ($\frac{\partial\wedge}{\partial D} = 0$, or differentiating Eq. (1) and setting the derivative equal to zero):

$$\frac{\partial\wedge}{\partial D} = -\frac{2\varnothing}{D^3} + k \sin\frac{\pi}{P}\,\rho \tan\alpha = 0 \qquad (2)$$

The length to diameter ratio of the rotor core $(L/D)_{min}$ giving the minimum steel and copper length $\wedge_{min}$ is



RADIAL AIR-GAP motor is rated at 3 hp, 1740 rpm. Observe that motor bearings are located under rotor fan blades. **(FIGURE 2)**

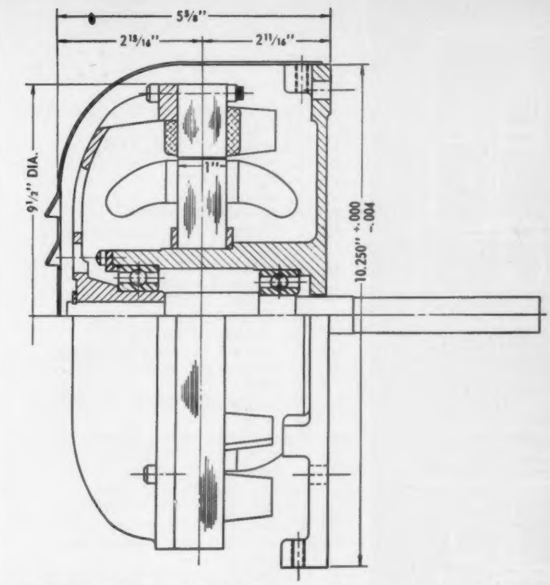
solved directly from Eq. (2) by directly substituting $\varnothing = D^2L$, which obtains

$$\left(\frac{L}{D}\right)_{\min} = \frac{k}{2} \sin \frac{\pi}{P} \rho \tan \alpha \qquad (3)$$

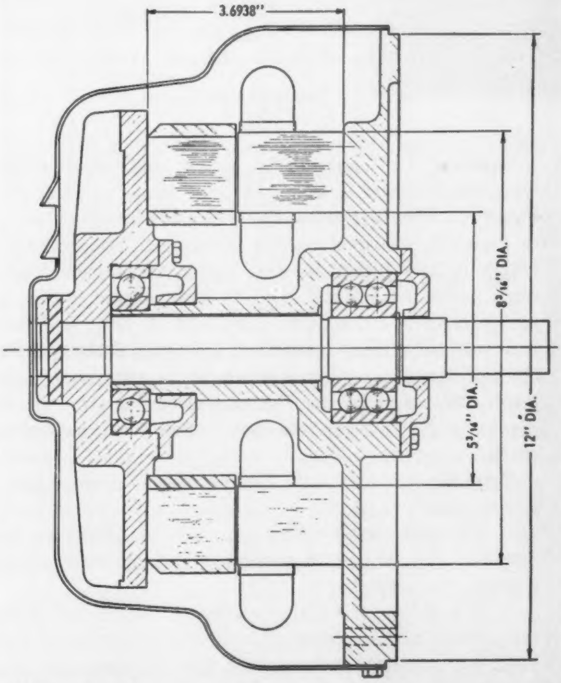
which is to say that above and below this limit the motor tends to be longer. Specific values of the limit $(L/D)_{\min}$ for the rotor core range more or less around 0.18 to 0.4, depending on the coil pitch, the number of poles, and so on. In this connection the actual steel and copper minimum length $\Lambda_{\min}$ obtained from Eqs. (1) and (2) is

$$\Lambda_{\min} = 2\beta + \frac{3}{4^{1/3}} \varnothing^{1/3} \left(k \sin \frac{\pi}{P} \rho \tan \alpha\right)^{2/3} \qquad (4)$$

Equation (4), plotted in Figure 8, is an important consequence. It indicates approximately how flat a radial air-gap pancake motor can be built. The equation deals with active parts; that is, the core and copper. The total length of the motor can then be determined. Curves similar to that shown by Figure 8, but fitting the specific practice of other designers and other manufacturing procedure, are easily constructed with the theory used to derive Eq. (1) and Eq. (2) combined to give Eq. (4). Other pole numbers and in instances where end extensions are mechanically compressed to further shorten them are easily incorporated.

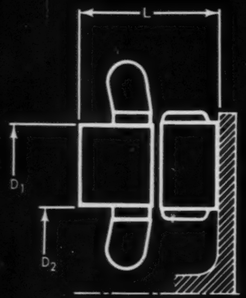


INVERTED RADIAL AIR-GAP motor is rated 1 hp, 1750 rpm. Coils are compressed to further reduce motor length. **(FIGURE 3)**

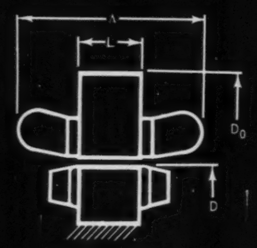


AXIAL AIR-GAP motor is rated at 2 hp and 3450 rpm. Design of motor components is best adapted to the flat shape. **(FIGURE 4)**

THREE-HP RADIAL air-gap motor's mechanical construction is similar to a conventional motor. **(FIGURE 5)**



AXIAL AIRGAP DIMENSIONS



RADIAL AIRGAP DIMENSIONS

Λ	= LENGTH OF COPPER AND STEEL PARTS
L	= LENGTH OF STEEL PARTS
$\varnothing$	= ROTOR D^2L
$\cos\theta$	= POWER FACTOR
η	= EFFICIENCY
P	= NUMBER OF POLES
k_{pd}	= PRODUCT OF PITCH AND DISTRIBUTION FACTOR
ϕ	= FLUX DENSITY
B_g	= AVERAGE AIRGAP DENSITY
q	= AMPERE COND., PER INCH
N	= SERIES TURNS PER PHASE
m	= NUMBER OF PHASES
V	= PRIMARY VOLTAGE PER PHASE
E	= SECONDARY VOLTAGE, EQUIVALENT
f	= FREQUENCY, CYCLES PER SECOND
I	= PRIMARY PHASE AMPERES
β	= COIL STICKOUT STRAIGHT PORTION (ONLY) INCHES
α	= COIL END TURN ANGLE
D	= ROTOR OUTSIDE DIAMETER, RADIAL AIR GAP
D_0	= STATOR OUTSIDE DIAMETER, RADIAL AIR GAP
D_1	= STATOR OUTSIDE DIAMETER, AXIAL AIR GAP
D_2	= STATOR INSIDE DIAMETER, AXIAL AIR GAP
kD	= AVERAGE DIAMETER OF END EXTENSIONS
ρ	= COIL PITCH EXPRESSED AS A FRACTION

THESE SYMBOLS used in text. (FIG. 6)

Equation (1) giving the copper and steel length spectrum L versus D of a particular motor is plotted in Figure 9. In this instance, the coil geometry of the motor in Figure 2 was used to plot the curve. The minimum length is 5.45 inches for this motor (the actual length came out to be 5.5 inches). Note in Figure 9 that, in general, motors having core diameters too large will have their overall lengths dominated by the geometry of the coil end extensions which is given by the asymptote on the right, and motors having too small core diameters will be dominated by the rotor core length given by the asymptote on the left.

From Eq. (3) and Eq. (4), the rules to achieve flatter motors are:

1. Use the correct $(L/D)_{min}$ ratio as given, for example, by Eq. (3) or an equivalent formula employing a cognate coil geometry.
2. Use the shortest coil extensions or, in the case given, the smallest angle α compatible with coil winding and assembly economies, and augment this by physically compressing the coils after assembly.
3. Use the shortest coil pitch ρ compatible with good motor performance.

Aside from these electrical considerations, much can be done in the way of mechanical design to obtain flatter motors. For example, Figure 2 shows a radial air-gap motor in which, to achieve flatness, the bearings are located directly under the rotor fan blades. The contour of the bearing housing at this point forms a natural air passageway, eliminating the usual air deflectors but providing sufficient ventilation over end extensions and core ends.

The manufacturing procedure which compresses the coils after they are assembled in the stator can be very effective in further reducing motor length.

The radial air-gap motor excels in mechanical construction, ease of maintenance, and durability. In particular, being similar to conventional motors, they are well understood and easily serviced by repair shops. There are no thrust loads imposed on the bearings, as in the axial air-gap motor, except those relatively minor thrust loads caused by the skewing of slots.

Stator is inside the rotor in inverted radial air-gap motor

The inverted radial air-gap motor differs from the conventional pancake motor in that the rotor revolves outside, instead of inside, the stator. Figures 3 and 10 show such a design. There are several applications, such as drives for large fans and for those requiring high Wk^2, for which this type of motor is naturally adaptable, excluding any consideration of size or shape. However, with what has been shown regarding the ordinary radial air-gap motor, it can be made flatter in the inverted form. Because the motor has a smaller end-extension diameter, the end extensions can be made shorter.

The analytical process described in the preceding paragraphs applies exactly to this motor. The value of the $(L/D)_{min}$ limit is approximately 30 percent less because k in Eq. (3) is about 30 percent less.

Theoretically axial air-gap motor is flattest

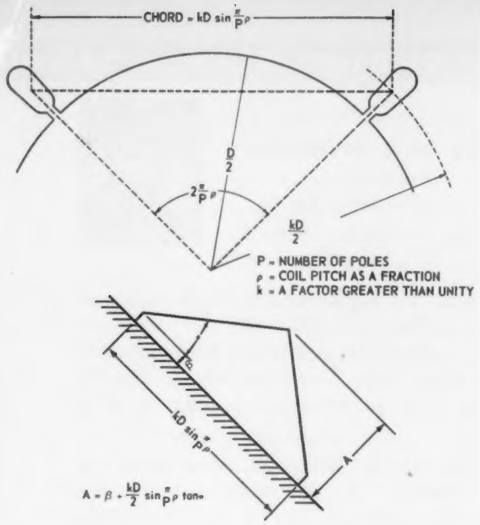
The axial air-gap motor, shown in Figure 4, represents considerable deviation from conventional construction, because the length of its air gap is measured in a direction parallel to the axis of the shaft.

The stator, shown in Figure 11, and rotor core of the axial air-gap motor are made from a continuous strip of silicon steel wound concentrically, like a spring. Slots are punched into this strip and spaced at increasing intervals so that the slots line up in the completed or wound-up core. The stator coil sides diverge like radii of circles, and the coil end length is large at the large radius end and small at the small radius end.

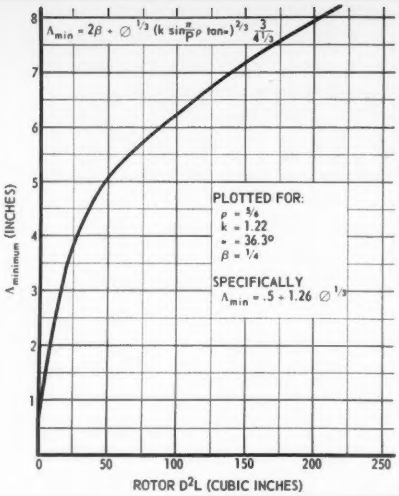
Unlike air gap in a radial air-gap motor, there is a magnetic force arising from the axial air-gap flux tending to close the air gap or to pull the rotor into the stator. The reactions of this force are borne by the bearings. Because of this force, the axial air-gap motor should have no end play to insure that the air gap, which influences motor performance, remains fixed.

Electrically, this motor is most adaptable, in fact practically restricted, to a pancake shape. The minimum length criteria for this motor are less essential, depend more upon detailed analysis of slot depth, core depth, and so forth, and permit a flatter motor in the realm of pure theory, while losing some of this advantage in the light of production requirements. The axial air-gap design excels from the standpoint of flatness, but it is more complicated than the radial air-gap design and provisions must be made to absorb inherent pull between rotor and stator.

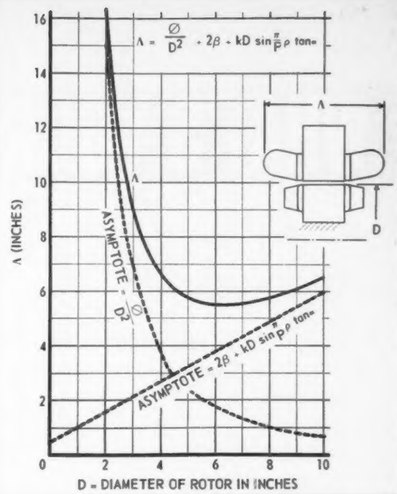
The special adaptability of the axial air-gap design to a flat shape is well illustrated by the "D_0^2L" formula which is a measure of the volume of steel in the motor. The symbolism is better written as $(D_1^2 - D_2^2)L$ where D_1, D_2 are the outer and inner diameters, respectively, of the axial air-gap stator or rotor and L is the combined



RADIAL air-gap motor end extension geometry shows how end extensions are developed. **(FIG. 7)**



CURVE plots the minimum length (copper and steel) of 4-pole radial air-gap motor. **(FIG. 8)**



CURVE plots length of active parts of radial air-gap motor versus core diameter. **(FIG. 9)**

length of the stator and rotor. The formula is obtained by solving four equations simultaneously. The first equation expresses the law of induced emf, the second multiplies average flux density B_g by the air-gap area $\frac{\pi}{4}\frac{(D_1^2 - D_2^2)}{P}$ to get flux per pole ϕ, the third equation is the basic expression relating horsepower output to kilowatt input, and the fourth equation defines q, that is, average ampere conductors per inch of circumference.

$$E = 4.44 f N k_{pd} \phi \times 10^{-8}$$

$$\phi = \frac{\pi}{4}\,\frac{D_1^2 - D_2^2}{P}\,\beta_g$$

$$HP = \frac{mVI\gamma\cos\theta}{746}$$

$$q = \frac{2mNI}{\pi D_{av}} \ldots\ldots\ldots D_{av} = \frac{D_1 + D_2}{2}$$

When solved simultaneously and multiplied by L the above four equations obtain for the axial air-gap motor the following equation:

$$(D_1^2 - D_2^2)L = \left[\frac{34HP \times P \times 10^8}{\beta_g\gamma\cos\theta\, qfk_{pd}\,\frac{V}{E}}\right]\frac{4L}{D_{av}} \qquad (5)$$

Equation (5), when multiplied by $\pi/4$, gives the volume of steel (cubic inches) of the combined rotor and stator. The factor $4L/D_{av}$ which multiplies the bracketed part of Eq. (5) clearly states that the steel volume becomes less as the ratio L/D_{av} becomes less or as the motor grows flatter. Thus the axial air-gap motor is practically restricted to a flat shape.

An interesting comparison emerges from Eq. (5). The bracketed part is exactly the D^2L equation for the rotor of a radial air-gap motor. This rotor D^2L also has been called $\varnothing$ when a definite fixed value was assigned. Thus with

$$\varnothing = \frac{34HP \times P \times 10^8}{\beta_g\gamma\cos\theta\, qfk_{pd}\,\frac{V}{E}},$$

the total volume of steel (except for a factor $\pi/4$) in the axial and radial air-gap designs is then:

$$(D_1^2 - D_2^2)L = \varnothing\,\frac{4L}{D_{av}} \ldots\ldots\ldots \text{axial air gap}$$

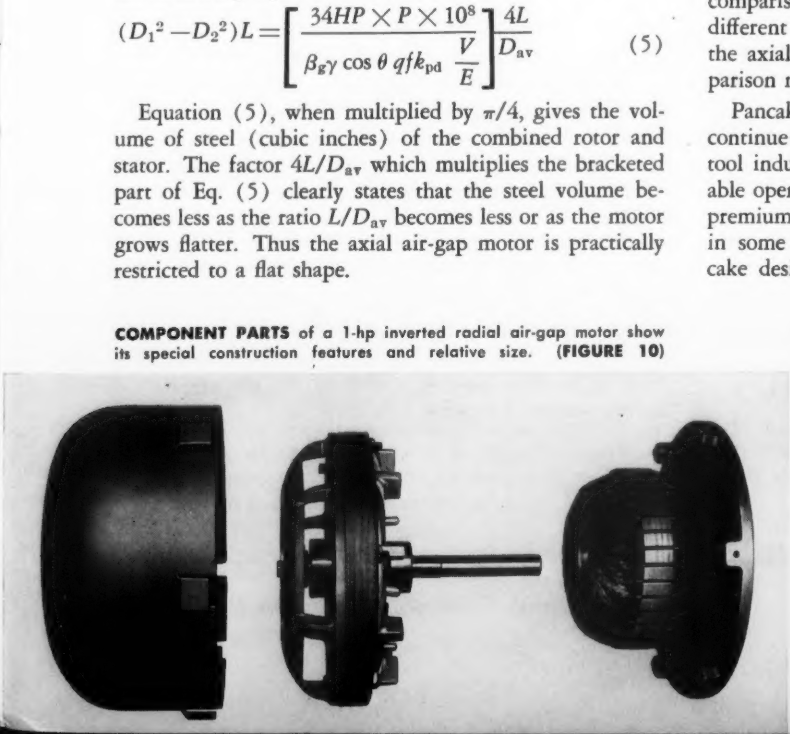
$$D_0^2L = \varnothing\left(\frac{D_0}{D}\right)^2 \ldots\ldots\ldots \text{radial air gap.}$$

Steel and copper parts decide motor shape

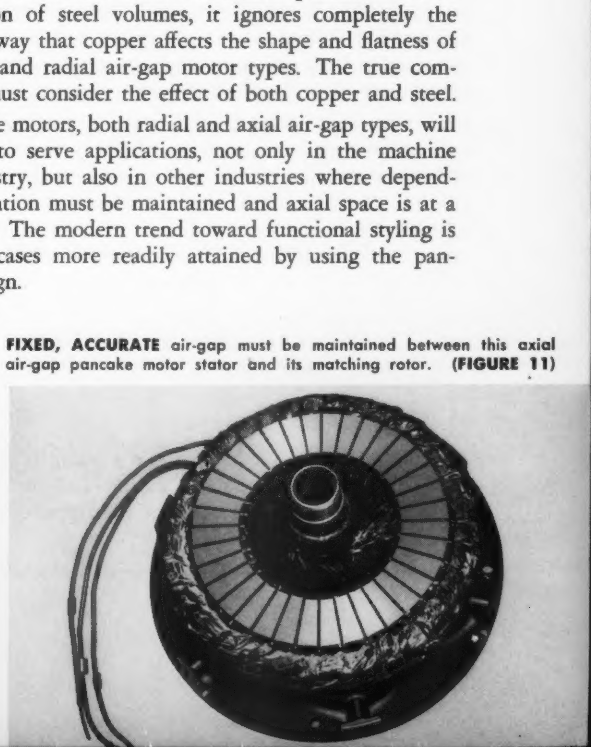
It is reasonable to expect that the combination of the group of factors comprising HP, poles β_g, f, q, and so forth, to be constant in order to allow direct comparison of steel volumes. While the above discussion permits a direct comparison of steel volumes, it ignores completely the different way that copper affects the shape and flatness of the axial and radial air-gap motor types. The true comparison must consider the effect of both copper and steel.

Pancake motors, both radial and axial air-gap types, will continue to serve applications, not only in the machine tool industry, but also in other industries where dependable operation must be maintained and axial space is at a premium. The modern trend toward functional styling is in some cases more readily attained by using the pancake design.

COMPONENT PARTS of a 1-hp inverted radial air-gap motor show its special construction features and relative size. **(FIGURE 10)**



FIXED, ACCURATE air-gap must be maintained between this axial air-gap pancake motor stator and its matching rotor. **(FIGURE 11)**



WHAT'S YOUR PATENT I.Q.

by **H. L. SWENSON**
Patent Attorney
Allis-Chalmers Mfg. Co.

A greater understanding of the government patent system will help an engineer protect his ideas.

ABRAHAM LINCOLN ONCE SAID, "The patent system added the fuel of interest to the fire of genius." As evidenced by the industrial success of our nation, our patent system has proven to be a durable fuel which has stimulated creative thinking into providing countless improved products. In stimulating investment of capital in research and new manufacturing enterprise, the patent system has become a dynamic force in maintaining our competitive economy — an essential element of our democratic government.

It is advantageous for engineers and others associated with the sciences to become familiar with the patent system, for it is an engineering tool which, when properly used, can be of great value. You must become familiar with the working of the patent system in order to use it proficiently.

Are you familiar with the philosophy upon which our patent system is founded? Are you aware of the benefits derived from owning a patent? Patents, like engineering reports, have a language all their own. Do you know the meanings of some of the terms most frequently used in relation to patents?

The following questions should help you in determining your patent I.Q. Even if you score low on this quiz, exposing yourself to the purposes, effects, structure and general background of the patent system will assist you in your use of it as an engineering tool.

QUESTION	ANSWER
1. Why do we have a patent system?	1. The patent system was created to promote the progress of science and useful arts. It does this because the protection afforded by the granted right to exclude others from making, using or selling, encourages people to make their inventions public knowledge. Further, it establishes a property right which can be sold or licensed by the inventor as desired.
2. The following is the heading on a patent: August 10, 1954 — J. B. Hodtum 2,686,236 Rotary Switch With Three-Point Contact Support Filed April 5, 1952	2.
(a) This patent expires on August 10, 1971 or April 5, 1969 or April 5, 1972?	(a) The patent expires on August 10, 1971, 17 years from its date of issue.
(b) The patentee may exclude others from making, using or selling his invention commencing on April 5, 1952 or August 10, 1954?	(b) Exclusive rights to make, use or sell the patented invention are granted on the date of the issue and not on the date the application was filed. The answer is August 10, 1954.
3. What is entitled to patent protection?	3. Any new and useful process, machine, manufacture or composition of matter or any new and useful improvement thereof.
4. Do you have to use an invention in order to get patent protection?	4. No. An invention need not be used in order to obtain patent protection. In fact you do not even have to build a working model of the invention before obtaining a patent.
5. Does it ever happen that a patent is granted and later it is decided that a mistake had been made and the patent should not have been granted?	5. Yes. If the patent is not granted to the proper person or if it does not in fact cover a patentably novel idea, it can be contested and declared invalid.

QUESTION	ANSWER
6. Can you get patents on a lot of ideas capable of embodiment which you are not going to use just to keep others from using them?	6. Yes; however, the fact that many companies do not use all of their patented inventions does not mean that they are suppressing the inventions. Frequently, the patented invention may lack immediate commercial value and, consequently, is not produced.
7. What does this mean? "Pat. Pend."	7. "Pat. Pend." means that a patent application is pending, or in other words, that an application for a patent has been filed.
8. Must a patentee mark all of his patented items with their corresponding numbers?	8. No; however, it is frequently done because the marking serves as a notice that the product is covered by one or more patents and may enhance the recovery of damages for infringement.
9. Why is there sometimes more than one patent number on a product?	9. A product may often contain several patentable inventions and, therefore, several patent numbers will be marked on the product. The entire product or only a portion of the product may be covered by one of the patents.
10. How much does it cost to file a patent application?	10. The government filing fee is $30.00. However, the cost of preparing the necessary papers and prosecution of the application may run from several hundred to several thousand dollars. The government final fee for issuance of the patent is an additional $30.00.
11. How long does it take before you know whether or not you are going to get a patent?	11. In two or three years you usually know whether or not you are going to get a patent. Most patents are issued anywhere from two to five years after the date of filing the application.

12. The following type of paragraph usually appears at the end of recently issued patents:

REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	*Name*	*Date*
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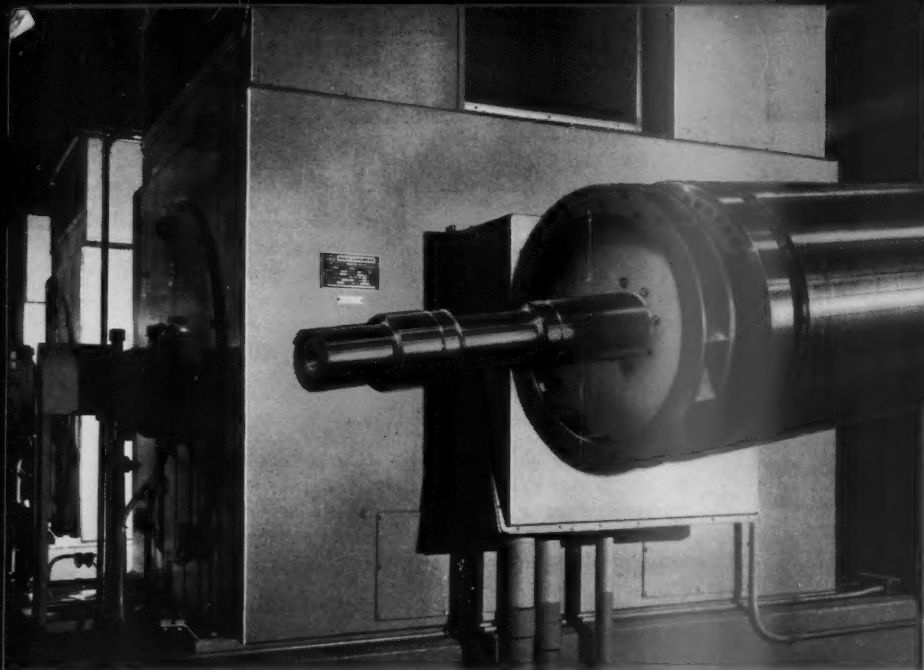
2,340,644	Clark	February 1, 1944
2,361,044	Mattox	October 24, 1944

OTHER REFERENCES

Dielectric Constant and Molecular Structure, 1931, pp. 197, 198 and 204, C. P. Smyth, Chem. Catalog Co., New York, N. Y.

The above list is:

QUESTION	ANSWER
(a) A list of others claiming the same invention?	
(b) A bibliography referred to by the patentee in his patent?	
(c) A list of material believed to be pertinent art which the Examiner in the U.S. Patent Office used during the prosecution of the application?	(c) This list is often helpful in finding prior art pertinent to the particular patent under consideration.



QUIET OPERATION of this 4000-hp, 3585-rpm, boiler-feed pump motor is essential. Careful balance of the motor's squirrel-cage rotor is an important factor in achieving this quietness.

BALANCING HEAVY SHAFTS and ROTORS

by **J. E. PETERMANN**
Motor and Generator Dept.
Allis-Chalmers Mfg. Co.

An understanding of static and dynamic balance problems using simple vectors takes mystery out of the art.

CONTINUAL GROWTH IN MODERN MACHINES is making balancing of heavy shafts and rotors a more important function in their manufacture. Normal balancing procedures are not generally difficult and the same general theory applies to all rotating equipment. Rotors of generators or large motors respond to the general theory whether they are horizontal or vertical.

The rotor to be balanced can be set up in its own bearings or in equivalent sized test bearings, and can be driven by a direct-current motor giving good speed control or by the method normal for the application.

The fundamental balancing is accomplished at full rotor operating speed with refinement at the critical speed or speeds, if criticals are a characteristic of the rotor involved. The critical speed occurs when the frequency of rotational speed coincides with the natural frequency of the shaft.

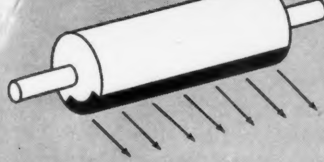
There are on the market approximately fifteen different types of instruments which give the amplitude and position of an unbalance. Normally with the readings obtained from these instruments a good balance can be obtained if a few basic rules are understood and observed.

Static and dynamic unbalance are terms frequently used in balancing discussions. Figure 1 illustrates the condition of static unbalance. In this case the rotor is heavy at the bottom, as shown by the dark area. If this rotor is supported in such a way that it can turn easily, it will always stop in the same position on coming to rest, thus proving that one side is heavy.

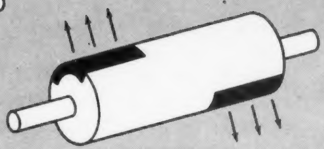
The condition of dynamic unbalance is illustrated in Figure 2. Equal amounts of unbalance are indicated by diametrically opposite dark sections. This rotor, if supported in such a way that it could turn easily, would stop in different positions in coming to rest, proving that it was statically balanced. However, if this rotor is revolving at high speed, the centrifugal force of these two unbalanced sections would act in opposite directions and create the unbalanced condition referred to as a dynamic unbalance.

Rules same for machines of any speed

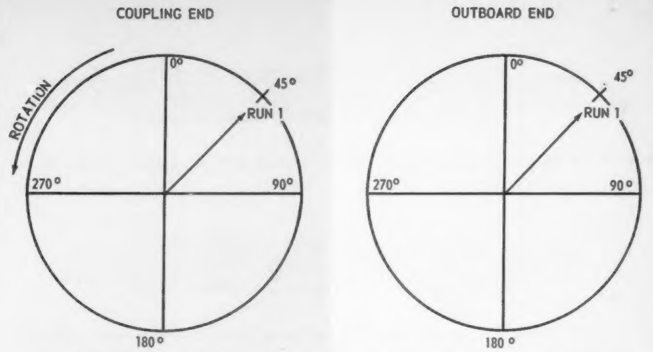
While a flexible rotor with normal operating speed of 3600 rpm is used as an example in the following discussion, the same basic rules apply to rotors for other operating speeds. The first balancing run for the driven rotor requires different handling than succeeding runs. The rotor is brought up to speed in 100-rpm steps until the highest reading of vibration on either pedestal is approximately 5 mils. Usually it will be necessary to limit the first run to some speed below 1000 rpm. A set of readings is taken on both pedestals, measuring the amplitude and location of both horizontal and vertical unbalance.



STATIC unbalance causes horizontal rotor to come to rest in same position. **(FIGURE 1)**



DYNAMIC unbalance is caused by two unbalanced components acting at higher speeds. **(FIG. 2)**



APPROXIMATE position and magnitude of static unbalance are determined on Run 1. Vector length equals four test-set divisions. **(FIGURE 3)**

The speed is carefully noted so that balancing for the next few runs can be carried on at the same speed. The unbalance showing up in these tests is predominantly static and is eliminated by using static weight.

Balancing at this speed is continued until the maximum amplitude at any point does not exceed 0.5 mil. In some cases static weight alone will not bring the amplitude of vibration down to this point, and dynamic weight must be added.

In the next step speed is increased until the balancer feels that the vibration has reached an amplitude which makes it inadvisable to go further. This speed, too, must be carefully noted, and balancing is continued at this point long enough to reduce the amplitude of vibration sufficiently so that speed may again be increased. The goal of the balancer, from this point on, is to get the rotor up to rated speed. Unbalance at the higher speeds is generally dynamic.

As soon as the rotor can be brought up to rated speed with a nominal amount of vibration, enough data should be taken to establish a resonance curve. This includes vibration readings on the bearing pedestals, measured and checked in the vertical, horizontal and axial direction for every 100 rpm up to rated speed. It is plotted on a curve sheet immediately and used as a guide for determining the speed or speeds at which to balance. With this data critical speed vibration is held within acceptable limits while doing balancing at full speed. A balancer must always evaluate the effects of balance weights for both full speed and the critical speed.

Typical static unbalance problem shows method

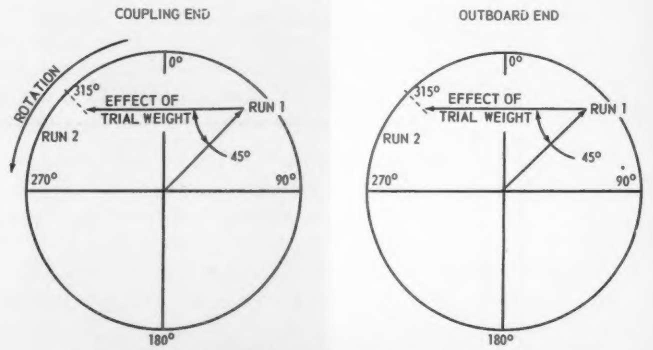
Static unbalance usually requires that the balancer consider only horizontal amplitudes and angles when balancing a horizontal rotor. In a typical case, the following vibration readings were taken with a balancing set on the first run:

Coupling end ....4.0 divisions at 45 degrees

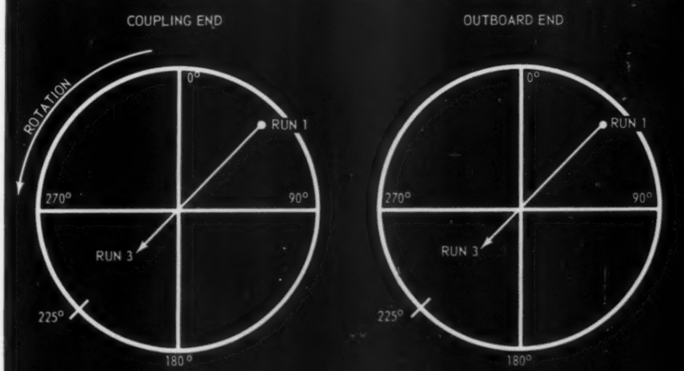
Outboard end ....4.0 divisions at 45 degrees

These values are plotted as vectors in Figure 3 and labeled Run 1. Although the actual value of divisions used depends on the particular balancing instrument, this fact does not alter the problem or its solution.

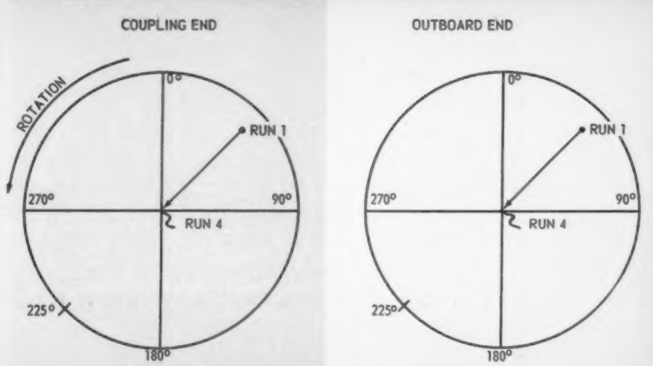
To correct the vibration detected in Run 1, a trial weight is placed in any one of the grouping of tapped holes at the mid-section of the rotor. The amount and position of the trial weight are based on the balancer's experience.



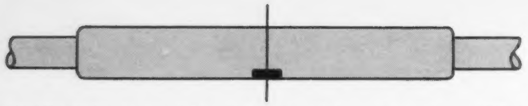
ANGULAR POSITION of trial weight in Run 2 is incorrect and balance is not improved. Weight was repositioned for Run 3. **(FIG. 4)**



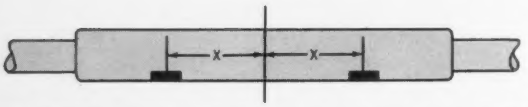
VECTOR LENGTH of Run 3 is 1.44 times the length of the Run 1 vector, indicating overcompensation by first trial weight. **(FIGURE 5)**



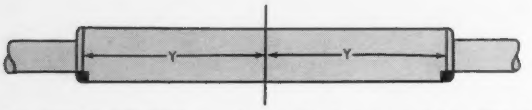
CORRECT WEIGHT in correct position returns vector for Run 4 to the center of the circle diagram for static balance. **(FIG. 6)**



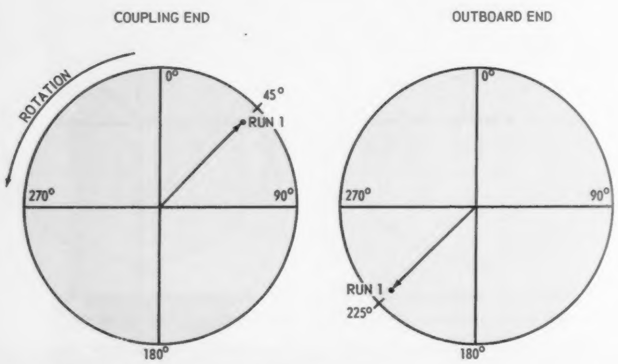
STATIC balancing weight can be placed in rotor center. **(FIG. 7)**



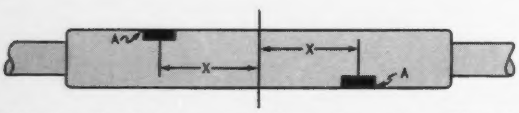
TWO WEIGHTS spaced equally from center serve same purpose. **(FIG. 8)**



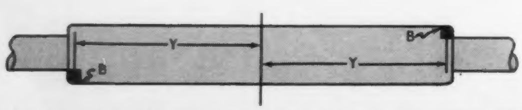
TWO STATIC WEIGHTS must be exactly the same weight. **(FIG. 9)**



DYNAMIC UNBALANCE is solved with same general method applying to static balance problems, but is more involved. **(FIGURE 10)**



TRIAL dynamic weights are same distance from center. **(FIG. 11)**



TRIAL dynamic weights may be shifted to the rotor ends. **(FIG. 12)**

Assume, for example, that a two-pound trial weight is placed at the center section of the rotor at 270 degrees. Balance is then checked at the same speed as before, and the following set of vibration readings are obtained with the balance set:

Coupling end ...4.0 divisions at 315 degrees
Outboard end ...4.0 divisions at 315 degrees

The points for Run 1 and Run 2 are plotted in Figure 4. The change between these points represents the effect of adding the two-pound weight at 270 degrees. It is readily apparent that the trial weight was added in the wrong position, and that the balance is not better than it was in the first place. Actually, when the rotor is in balance, the plot of readings obtained will be very near the centers of the circles.

Now that the effect of the trial weight is known, Figure 4 shows that the change obtained is 45 degrees away from bringing the plot of readings toward the center. To correct the vector position the weight is moved 45 degrees in the direction of rotation to position 225 degrees.

The following results would then be obtained:

Coupling end ...1.7 divisions at 225 degrees
Outboard end ...1.7 divisions at 225 degrees

Figure 5 shows that the balance weight is now located in the correct position, but that too much weight has been added. In the illustration, the vector change from Run 1 to Run 3 is 1.44 times Run 1 vector length, the result of adding two pounds of weight. To completely correct this unbalance the vector change should have been 1.0 times the length of Run 1 vector, which would locate the final plot exactly at the center of the circle.

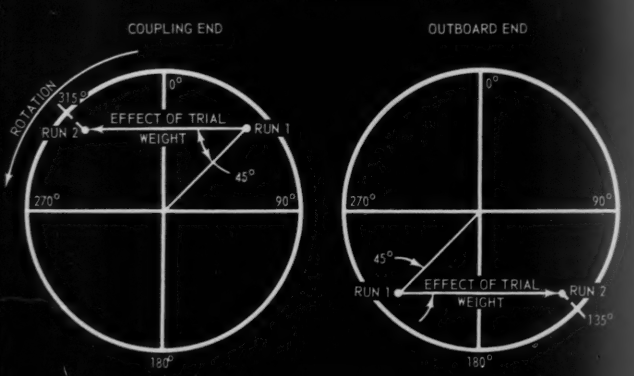
The correct weight to be added at 225 degrees can be determined from the facts already known. It is in the same proportion to the trial weight as the vector for Run 1, as based on original unbalance, to the vector for Run 3, as based on unbalance after adding two-pound weight in the corrected position 225 degrees.

This could be expressed in the following formula:

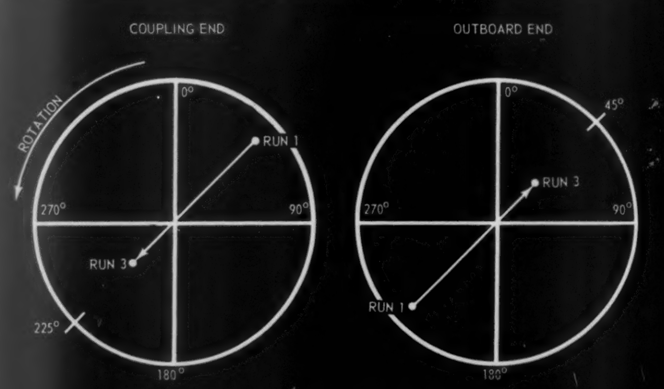
$$\frac{\text{Run 1 vector}}{\text{Run 3 vector}} \times \text{trial weight} = \text{correct weight}$$

or, in this instance,

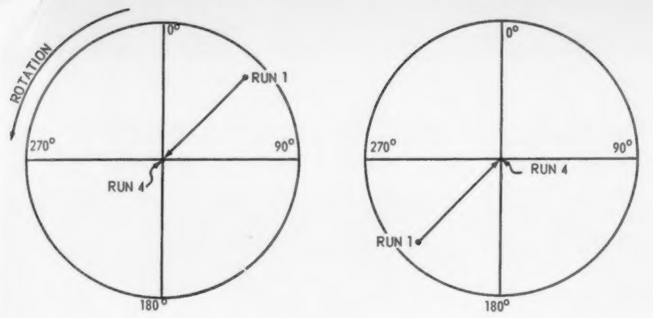
$$\frac{1.0}{1.44} \times 2 = 1.39 \text{ (pounds correct weight)}$$



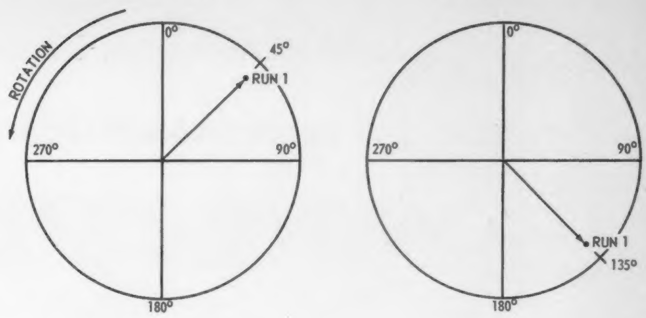
EFFECT OF TRIAL WEIGHTS is demonstrated in Run 2. Both trial weights are in wrong position and balance is not improved. **(FIGURE 13)**



TRIAL WEIGHTS are moved in the direction of rotation for Run 3 to offset the error in positioning of Run 2. **(FIG. 14)**



VECTOR of Run 3 overcompensates for the unbalance by 1.44 times. Corrected weight of Run 4 provides proper balance. **(FIGURE 15)**



COMBINATION static and dynamic unbalance appears in Run 1. Problem is simplified by considering static component first. **(FIG. 16)**

The result of adding the correct weight in the correct position is shown in Figure 6.

This typical case illustrates the procedure for taking care of a static unbalance. The suggested position for adding weight was at the center, or mid-section, of the rotor (Figure 7).

Other methods of adding static weight to a rotor are shown in Figures 8 and 9. Figure 8 shows the addition of static weight at intermediate points and Figure 9 shows the addition of static weight to the ends of the rotor.

When adding static weight as shown in Figures 8 and 9, it must always be added in two parts to obtain a pure static change. It is important that the two parts be of exactly the same weight. Further, additions must always be in the same circumferential location and the same relative axial position, as shown in the illustrations.

The typical case as presented, and illustrated in Figures 3 through 6, holds true for any of the instances of weight placement, Figures 7 through 9.

Typical dynamic unbalance problem solved

Although the solution to a condition of dynamic unbalance is somewhat more complicated, the approach is similar to that outlined for static unbalance. Considering a typical case, vibration readings taken with a balance set on the first run were as follows:

Coupling end . . . 4.0 divisions at 45 degrees

Outboard end . . . 4.0 divisions at 225 degrees

These values are plotted in Figure 10 and are labeled Run 1. To correct the unbalance which has caused this vibration it will be necessary to make a trial dynamic weight change, as shown in Figures 11 and 12.

Figure 11 represents a dynamic weight change at an intermediate point of the rotor. The weights "A" are equal and are located the same distance X from the center section of the rotor. Circumferentially, they are located 180 degrees apart. These conditions must be fulfilled exactly to call a weight change a dynamic weight change.

Figure 12 represents a dynamic weight change in the ends of the rotor. The weights "B" are equal and are located the same distance Y from the center section of the rotor. Circumferentially, they are exactly 180 degrees apart.

Suppose, for example, that a one-pound trial dynamic weight is placed at 315 degrees on the coupling end and that a similar one-pound weight is placed at 135 degrees on the outboard end. The results obtained from adding these weights are shown in Figure 13.

Points for Run 1 and Run 2 are plotted on the diagrams in Figure 13. The change between these points represents the effect of the trial dynamic weight which has been added. Since, as before, the plot of readings for a balanced rotor would be near the centers of the circles, it is apparent that the trial weight was added in the wrong position.

Now that the effect of the trial weight is known, Figure 13 shows that the change obtained is in a direction 45 degrees away from bringing the readings toward the center. To correct the weight position and balance the rotor, the weight added as a trial would have to be moved 45 degrees in the direction of rotation on both ends. This would bring the weight on the coupling end to 270 degrees and on the outboard end to 90 degrees.

When run with weights in the new positions the rotor would show the following vibration readings:

Coupling end . . 1.75 divisions at 225 degrees

Outboard end . . 1.75 divisions at 45 degrees

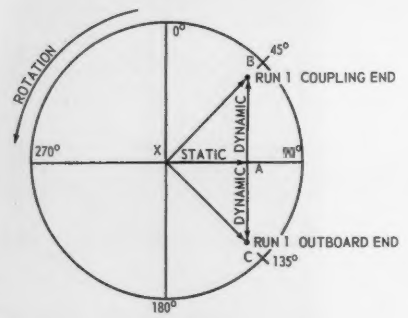
These values are plotted in Figure 14 and are labeled Run 3.

It is evident in Figure 14 that the balance weight is located in the correct position, but that too much weight has been added. The vector change here from Run 1 to Run 3 is 1.44 times the length of Run 1 vector, the result of adding one pound of dynamic weight. In order to finish correcting this unbalance the vector change should have been 1.0 times the length of Run 1 vector.

Using this data in the formula previously referred to, the correct weight to be added can readily be determined:

$$\frac{1.0}{1.44} \times 1 = 0.695 \text{ pounds}$$

Note that where one pound of dynamic weight is referred to, it means that one pound of weight has been added to each end of the rotor, and that these two weights are diametrically opposite.



SUPERIMPOSING coupling and outboard vector diagrams of Run 1 solves for the static component as well as dynamic component. (FIG. 17)

Figure 15 shows the result of adding the correct weight (0.695 pounds) in the correct positions (270 degrees on the coupling end and 90 degrees on the outboard end).

Typical case of static and dynamic unbalance

Having considered cases where unbalance can be simply defined as either static or dynamic, it is well to remember that balance problems are not always so limited. In a typical case where both static and dynamic unbalance were discovered, the following vibration readings were taken:

Coupling end . . . 4.0 divisions at 45 degrees
Outboard end . . . 4.0 divisions at 135 degrees

These values are plotted in Figure 16 and labeled Run 1.

There are two methods by which the unbalance which has caused this vibration can be corrected:

(1) Eliminate static and dynamic components separately.

(2) Eliminate static and dynamic components by using a combination of static and dynamic weight.

Because it is simpler to analyze, the first method should be used, and static and dynamic components should be separately handled. Experience has proved that, while it may require more balancing time, this method gives the best results.

In Figure 17 the horizontal readings of both bearings have been plotted on one circle. By this means the vibration can be analyzed to the extent that the static and dynamic components can be separately determined.

The construction in Figure 17 is obtained by first connecting points *B* and *C* and then drawing a line from *X* to *A*, the mid-point of line *BC*. Line *XA* now represents the static component of unbalance for both bearings, line *AB* is the dynamic component for the coupling end bearing, and line *AC* is the dynamic component for the outboard bearing.

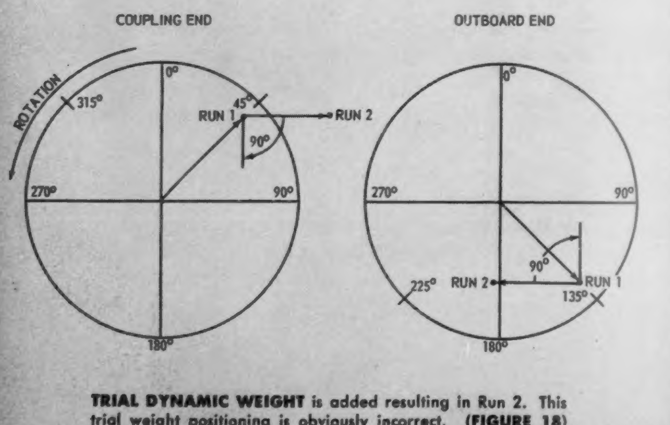
For the first trial run in solving this problem a dynamic weight change is made to add one pound of weight at 315 degrees on the coupling end disk and an equal weight at 135 degrees to the disk on the outboard end. Figure 18 shows the results obtained from these weight additions.

Points for Run 1 and Run 2 are plotted in Figure 18. The change between points represents the effect of the trial dynamic weight which has been added. Since the purpose of putting this dynamic weight in the rotor was to bring the points for Run 2 together, leaving static unbalance, this first trial was obviously not correct.

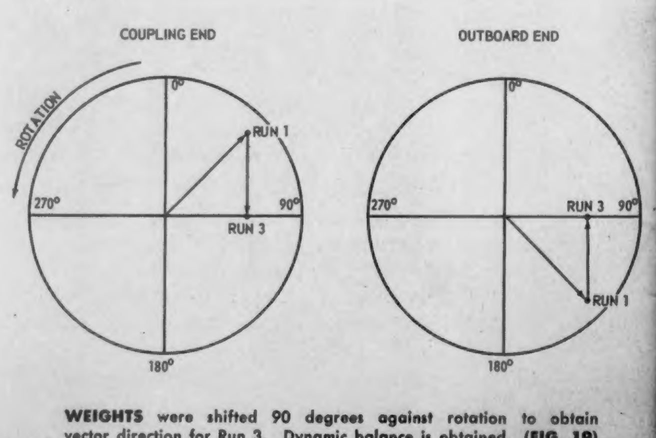
The positions of weights will have to be moved 90 degrees against rotation to get the vector change in the direction desired. This means that the one-pound weights will have to be moved from 315 degrees to 45 degrees on the coupling end, and from 135 degrees to 225 degrees on the outboard end. The results obtained from changing the positions of these weights are shown in Figure 19.

The plot in Figure 19 for Run 3 shows that the dynamic component of unbalance has been eliminated, since identical readings (2.87 at 90 degrees) have been obtained on both bearings. The remaining static unbalance can now be eliminated by following the procedure outlined in the first example — that of static unbalance. The case used in the example involves a flexible rotor which is long and slender and has a critical speed below its operating speed. A stiff rotor is one which normally does not run through a critical before reaching operating speed and can often be balanced by using only static weights, thus making the balancing operation extremely simple.

Observation of the points emphasized in this basic procedure should allow anyone to do a good job of balancing with a nominal number of trial runs. The degree of success attained will depend on the extent to which these simple concepts are followed.



TRIAL DYNAMIC WEIGHT is added resulting in Run 2. This trial weight positioning is obviously incorrect. (FIGURE 18)



WEIGHTS were shifted 90 degrees against rotation to obtain vector direction for Run 3. Dynamic balance is obtained. (FIG. 19)